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16. Abstract

   The state of the art in simulating the dynamic behavior of road vehicles is summarized and then applied to a specific military vehicle (the M-151) to determine the operating conditions and maneuvers that are likely to cause rollover. The findings obtained by means of simulation confirm and explain the rollover experience of the M-151 and also suggest why drivers of the M-151 may inadvertently bring the vehicle to an operating state in which the potential for rollover is large, given a need for some emergency steering or braking control. The simulation findings are examined to determine the feasibility of synthesizing a rollover index for use either as (1) a guide in assessing the suitability of a given vehicle for a given mission or (2) providing a warning to drivers such that they can avoid operating states promoting a rollover response.

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AN INVESTIGATION OF THE ROLLOVER DYNAMICS
OF A MILITARY VEHICLE

Final Report
P.O. 00204

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July 3, 1979

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1.0 INTRODUCTION

This report summarizes the findings of a study performed by the Highway Safety Research Institute (HSRI) of The University of Michigan for Varigas Research, Inc. The study was conducted as part of Task 1 of a project entitled "Military Vehicle Rollover Analysis and Instrumentation" in which HSRI was asked to define the problems encountered in calculating the potential for rollover possessed by pneumatic-tired military vehicles.

To attain this objective, HSRI and Varigas Research concluded that HSRI should conduct the following tasks:

1) summarize the state of the art in simulating the dynamic behavior of road vehicles,

2) apply this art to produce a mathematical description of a representative military vehicle reputed to have a rollover problem (namely, the M-151),

3) investigate the dynamic behavior of the M-151 to determine the operating conditions and maneuvers that are most likely to challenge its rollover immunity,

4) determine the rollover limits of the M-151 and contrast the roll behavior of the M-151 with that exhibited by representative passenger vehicles,

5) investigate the influence of load and load distribution on the rollover behavior of the M-151, and

6) analyze and interpret the above findings to determine (a) the feasibility of "synthesizing a rollover index" and/or (b) a combination of measurable input and output variables which indicate that rollover is likely to occur.

It should be emphasized at the outset that this study considers only the rollover phenomenon which occurs as a result of maneuvers performed by motor vehicles on paved, level surfaces. Clearly, military vehicles can encounter rollover-promoting conditions when they operate off road or when they inadvertently slide off a paved road surface.

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(Some of these rollovers, such as may occur when a vehicle attempts to negotiate too steep a sideslope, are amenable to a static analysis, whereas other "off road" rollovers, as caused by "skidding" from a pavement onto a soft shoulder or striking a curb, are difficult to categorize and interpret because of the infinity of conditions that can cause these kinds of rollovers to occur.) On the other hand, there is reason to believe that some of the military vehicles used by the U.S. Army are experiencing rollover incidents more frequently than other vehicles purely as a result of being driven at speed over a paved road network. In the context of this experience, it is possible to justify an examination of whether a given vehicle design (or in-use configuration) can experience a rollover event on a paved surface as a result of driver control actions. Although such an examination is most expeditiously performed by conducting a systematic series of full-scale tests on the vehicle in question, it can also be performed by means of simulation.

It should be noted that the latter methodology was employed in this study. This means that the findings have a certain level of uncertainty as derives from (1) inaccuracies in the data acquired or estimated to define a given vehicle and (2) the completeness and validity of the computer code that describes the dynamics of a tire-vehicle system. Given, however, the advantages and disadvantages of full-scale test and simulation, respectively, it was concluded that simulation constitutes the preferred methodology in light of the objectives of this study. We do not, however, want the reader to infer that simulation is the most cost-effective methodology under all circumstances and for all possible study objectives.

Section 2.0 of this report summarizes the state of art applicable to the simulation of the dynamic behavior of road vehicles, in general, and the prediction of rollover thresholds, in particular. It also discusses the rollover process and identifies the various properties of the motor vehicle which have a major influence on rolling behavior.

Section 3.0 identifies the computer code used to simulate the behavior of the M-151 in roll-provoking maneuvers and outlines the methods

employed and the assumptions that were made in establishing the data
and design parameters defining the M-151. Section 4.0 discusses the
dynamic behavior of the M-151, as predicted by the simulation employed
in this study for various combinations of steering, braking, and
acceleration inputs as a means of exploring the rollover potential of
the M-151. Study findings related to (1) defining the rollover threshold
of the M-151, (2) examining the influence of loading on this threshold,
and (3) comparing the behavior of the M-151 with a representative
passenger car are presented in Section 5.0. The prospects for synthe-
sizing a rollover index and recommendations for follow-on work are
presented in Sections 6.0 and 7.0, respectively. The references cited
in the text are listed in Section 8.0.

2.0 ASSESSMENT OF ROLLOVER POTENTIAL: THE
STATE OF THE ART

The mechanics of rollover, as can occur on a level, smooth surface, are discussed below (in very gross terms) prior to discussing and defining the present state of the art for assessing the rollover potential of a motor vehicle from both an analytical and/or experimental point of view. Although the emphasis is on predicting the rollover threshold applicable to maneuvers performed on a level, smooth surface, it will be argued that experimental methods of assessing rollover immunity (or rollover potential) can be more cost effective than simulation, if the vehicle exists as a physical entity. On the other hand, if the vehicle is only a proposed design, then analysis and simulation constitute the only method for determining the rollover threshold of a vehicle performing maneuvers on a level, smooth surface.

2.1 The Vehicle Rollover Process

The rollover potential of the military vehicle has always been a matter of concern, since these vehicles may, on occasion, be forced to traverse a side slope. In this instance, the vehicle can become statically unstable in roll purely as a result of gravitational forces causing a roll moment which exceeds the roll-resisting moments created at tire-road contact. No accelerations due to maneuvering are necessary; a standing vehicle will roll over if the gradient of the side slope is such that the gravitational force vector falls outside the track width of the vehicle.

On the other hand, a maneuvering vehicle can, in theory at least, roll or pitch over in a turning or braking maneuver. The potential for roll- or pitch-over is, of course, considerably increased if the running gear of the vehicle should encounter some obstacle (for example, a curb or rut in a soft shoulder) which can create a force tangential to ground which is considerably larger than the shear force which can be created by a frictional process. Accordingly, situations can arise in which a vehicle sliding on a hard, smooth surface encounters an obstacle (and

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possibly a slope as well) such that the "impact" forces occurring at
the running gear, together with the gravitational moments created by
a tilted surface, create moments sufficient to overturn the vehicle.
Clearly, the higher the center of mass above the supporting surface and
the shorter the wheelbase or track of the vehicle, the greater will be
the potential for overturning if a vehicle should slide and then "stub
its toe."

A more elusive overturning scenario than those mentioned above is
the case in which a turning vehicle (or a vehicle which is both decel-
erating and turning) rolls over on a level, smooth surface without en-·
countering (a) an obstacle or (b) a sudden increase in tire-road friction,
for example. The term "elusive" is used because a static analysis of
the mechanics involved is unable to indicate whether a given maneuver
will cause a rollover event. Rather, it is necessary to calculate the
directional response to steering and braking control inputs to determine
whether a rollover event will occur.

It should be noted that rollovers will not occur in maneuvers per-
formed on a level, smooth, and constant-friction surface when a driver
steers and brakes in a normal manner, namely, to accomplish typical
path- and speed-keeping objectives. Only when a driver finds it
necessary to perform an emergency maneuver, that is, he steers and
brakes so as to approach the maximum forces that can be generated by
the tires, is it possible or likely that a rollover threshold will be
exceeded.

For many years, however, motor vehicle engineers have tended to
believe that a motor vehicle having the size and shape of a representative
motor car would never exceed its rollover threshold on a smooth, level
surface, irrespective of the maneuver called for by the driver. Whereas
this belief is true for many motor cars, it is not true for all cars.
Further, experience has demonstrated that the payload-carrying objective
of the truck (or commercial vehicle) is such that rollover will, in gen-
eral, occur before the limit turning capability (as determined by tire-
road friction) is achieved. The logical questions to raise are "Why do
some motor cars rollover in maneuvers performed on a level, smooth sur-
face, when most do not?" and "Given that trucks have less immunity to

rollover than the typical motor car, does the motor truck have any directional response characteristics which might increase its potential for rollover?"

To address these questions, we should first consider what a simple static analysis tells us about the likelihood of rollover. Consider, first, a vehicle without a suspension sitting on tires whose radial stiffness is infinite. If we look at this vehicle from the rear (Fig. 1), we see that, in a steady left turn, the centripetal acceleration creates a centrifugal force that is equal and opposite to the side forces generated by the right and left tires. Equilibrium requires that the summation of the moments be zero, yielding that

$$F_{zR} = \frac{W}{2} + \frac{h}{t}(F_{yL} + F_{yR})$$

$$F_{zL} = \frac{W}{2} - \frac{h}{t}(F_{yL} + F_{yR})$$

where

    h = height of the center of mass of the total vehicle above the road

    t = track width (or tread)

    W = weight of the vehicle

    F_{yL}, F_{yR} = side forces acting on the left and right wheels, respectively

    F_{zL}, F_{zR} = normal forces acting on the left and right tires, respectively

It is seen that the normal load on the left (i.e., "inside") tire will disappear when

$$F_{yL} + F_{yR} = \frac{t/2}{h} W \qquad (1)$$

If it is assumed that the maximum side force which can be generated by a tire is given by
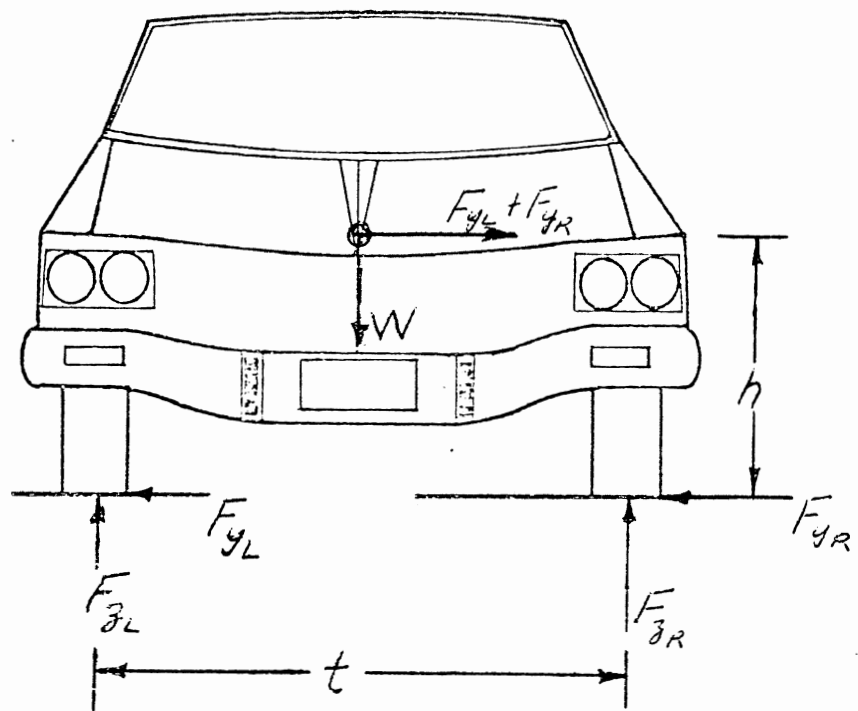
$$F_y)_{max} = \mu F_z$$

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Figure 1. View of a turning vehicle from the rear.

where

μ = tire-road friction coefficient,

we have that

$$(F_{yL} + F_{yR})_{max} = \mu W$$

Further,

$$F_{yL} + F_{yR} = \frac{a_y}{g} W \qquad (2)$$

where a_y/g = lateral acceleration (nondimensional).

On substituting Equation (2) into (1), we find that the lateral acceleration, a_y/g, in g units required to reduce the load on the inside wheel (in a steady turn) to zero is

$$\frac{a_y}{g} = \frac{t/2}{h}$$

For a typical motor car, t = 60 inches and h = 22 inches. Thus, the value of a_y/g required to reduce the load on the inside wheel to zero is 1.36. For a typical truck t = 80 inches and 50 in.< h < 80 inches. If h = 55 inches, a value of a_y/g = .73 will be sufficient to unload the inside wheel completely. Note that, if the vehicle has a suspension and the tires deflect radially and laterally, the resulting lateral shift of the center of mass unloads the inside wheel at a limit turn condition corresponding to centripetal accelerations which are somewhat lower than the values computed above. A static analysis of this kind indicates that motor cars (whose tires yield tire-road friction coefficients of μ < 1.0) should reach a limit turning capability that is substantially below that required to unload the inside wheels. Further, this same static analysis indicates that motor trucks are likely to approach and exceed their rollover threshold prior to attaining their limit turning capability, as defined in this very simple treatment of the turning vehicle.

The above calculations are very approximate and can be very misleading. First, they imply that, if rollover is ignored, the limit turn, in g units, will be equivalent to the prevailing tire-road friction coefficient. Both analysis and test show that this is not the case. A typical motor car is not able to "corner" at more than about 0.75 g on a surface which exhibits a tire-road friction coefficient in the neighborhood of 1.0. (A comparable experimental finding cannot be obtained for the motor truck, since the truck rolls over before its tires produce maximum side force.) Second, the calculation supports the conclusion that typical motor cars (i.e., passenger vehicles with a low c.g. height and a relatively wide track) do not roll over. This conclusion is not valid in all cases, because of two factors:

1) There are dynamic conditions created by time-varying steering inputs and by certain combinations of steering and braking which can produce upsetting moments sufficient to rollover some motor cars.

2) The rollover potential of a motor car is also significantly influenced by suspension geometry and, in particular, is influenced by the <u>kinematic</u> properties of its suspension, as viewed in the transverse plane, namely, the plane normal to the longitudinal axis of the vehicle.

For example, both analysis and experiment have shown that an independent suspension with a geometry yielding a high roll center leads to significant "jacking" forces at high g levels when the side forces on the tires are very asymmetrical, right to left. This phenomenon is a process with positive feedback, since a "jacking" force leads to a reduction in track width and a further increase in roll center height which, in turn, increases the "jacking" force and thereby reinforces the process, a process which could be described as a "suspension instability." A "suspension instability" of this kind is highly nonlinear and tends to develop only after a certain cornering threshold has been exceeded. However, the onset of "suspension instability" can lead to roll instability, as the c.g. of the vehicle rises and the track width of the unstable suspension is reduced.

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The question thus arises as to why a vehicle developer would use an independent suspension design for which the possibility of a "suspension instability" exists. The answer is multifaceted and complex. The problem generally arises when the designer wishes to give a vehicle an independent rear suspension and also employ rear-wheel drive. Design tradeoffs arise in which simplicity and cost are weighed against the undesirable features of an independent suspension with a high roll center. Since these undesirable features become apparent only when the vehicle is pushed to its cornering limit, they often are overlooked, or, if the designer is fully aware of these shortcomings, a judgment can be made that the "good" features outweigh the "bad." Whereas in the U.S., there has been only one rear-drive motor car built with an independent rear suspension possessing a high roll center (namely, the Chevrolet Corvair), such designs have not been uncommon in Europe. We also see this arrangement in vehicles designed for off-road use, such as the M-151. In general, accident records indicate that vehicles of this type are nearly always overinvolved in accidents which involve a rollover incident.

To conclude this discussion of the rollover process, it must be emphasized that, notwithstanding the attraction of analyzing rollover as a process in which statics are predominant, rollover is essentially a dynamic phenomenon. Recognition of this fact leads to the question as to whether there is a particular maneuvering sequence that imposes a maximal challenge to the rollover immunity of a motor vehicle. This question has been addressed by HSRI [1], using an experimental approach together with physical reasoning based on an understanding of the mechanics involved. In general, HSRI has found that the most demanding maneuver is a combined braking and steering maneuver that could occur in an obstacle-avoidance scenario in which the driver first steers, then brakes sufficiently to lock all wheels, and then releases the brake when he feels the vehicle beginning to slide sidewards. Application of this control sequence to a representative sample of motor cars has shown [2] that many motor cars cannot be rolled over on a level, smooth surface under any circumstances. However, some can, and tests [2] have also shown that some cars will rollover even when they are given only a

sudden, large steering input, as required to perform a limit J-turn maneuver. In addition, an HSRI staff member has seen movies of tests made in Japan in which a steer input intended to produce a severe lane-change maneuver is sufficient to cause rollover of the Japanese cars under test. Clearly, when the c.g.-height-to-track-width ratio is sufficiently large (as can occur in compact cars with a narrow track), a severe dynamic maneuver can cause cars to rollover even in the case of vehicles which do not exhibit any form of "suspension instability."

As indicated earlier, the above remarks do not apply to fully laden heavy trucks. Whereas a dynamic maneuver can precipitate a rollover at a g level which is less than that required to roll the truck in a steady turn, it is also true that almost any heavy-duty commercial vehicle, when fully laden, will rollover prior to reaching its limit steady-cornering capability. A discussion of the design features and variables which influence car and truck rollover under dynamic maneuvering conditions will be presented after first reviewing the state of the art in modeling tire-vehicle systems.

2.2   Models of Tire-Vehicle Systems

An analysis of the directional stability of the four-wheeled motor vehicle first appeared in the technical literature in 1940 [3], whereas analyses of the ride dynamics of the motor vehicle go back considerably further in time. The primary reason for this difference is that ride phenomena can be analyzed in terms of the behavior of simple mass-spring-damper systems that are easy to visualize in terms of the construction of the motor vehicle, whereas the analysis of the directional response to steering required an understanding of the process by which the pneumatic tire produces a side force. This understanding did not exist until the early thirties, when Broulhiet [4] first discussed (in 1925) the role of sideslip in the generation of side force, and researchers [5] in Germany subsequently performed what where (presumably) the first measurements ever made of the cornering stiffness of the pneumatic tire.

Subsequent to these early efforts (as made to develop an understanding of the directional dynamics of the motor car), a continuing sequence of analytical and experimental endeavors has taken place with

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the objective of increasing and improving our understanding of why and how the motor vehicle behaves as it does. In the time frame subsequent to the ending of World War II but prior to the ready availability of analog computers (mid-1950's), analytical efforts were restricted to the development of linear equations of motion which are most adequate for describing the behavior of a constant-speed vehicle conducting maneuvers which constitute a small disturbance from straight-line motion. The four-wheeled motor car was the exclusive object of attention and almost all of the studies addressed the rear-drive vehicle which had an independent front suspension and a solid axle at the rear. These linear analyses led to closed-form solutions, which solutions provided a clear understanding of the manner in which steering gain is affected by understeer and the manner in which yawing, sideslipping, and rolling response to steering is influenced by the linear understeer gradient and other design properties of the motor car [6].

The commercial availability of the electronic differential analyzer (more commonly known as the analog computer) in the mid-50's removed the requirement to linearize the equations describing the dynamics of the motor car. Among the various efforts made to exploit this new computer technology, the equations developed by Pacejka [7] and Bergman, et al. [8] stand out as pioneering contributions. However, the newly-developed ability to solve these equations did not prove to be very useful or productive in that the procedure poses a demanding requirement for information describing the inertial, mechanical, and geometric properties of the various components of the vehicle system. In particular, a requirement arises for describing the mechanical characteristics of tires in far greater detail than was typically available at that point in time. Thus, the radical improvement in the analyst's ability to treat complex, nonlinear mechanical systems created a need for descriptive data that cannot be provided unless (1) substantial measurements are performed in the laboratory and/or (2) additional calculations are conducted on the basis of information available in design drawings.

One should differentiate between an ability to model the motor vehicle and the generation of findings and understanding. Generally

speaking, the modeling endeavor is constrained by the capabilities and capacity of the computer that the analyst has at his disposal and/or the funds that are available for his study. As soon as the digital computer became generally available, the constraints on the modeler changed in a drastic manner in that it now became possible to model a vehicle system as completely as desired as long as computational costs are not an overriding consideration.

Under this changed environment, a requirement arose for predicting the trajectory of the motor car when leaving the road or after impacting a median barrier, for example. Under the auspices of the Federal Highway Administration, a substantial effort was mounted at the Cornell Aeronautical Laboratory (now Calspan, Inc.) to develop a digital computer code providing the desired capability. This code, known as the "Highway-Vehicle-Object Simulation Model" (HVOSM), is applicable only to a four-wheeled vehicle but, on the other hand, is essentially unrestricted with respect to the motions or trajectories that can be accommodated. The guiding philosophy used in developing this code was to obtain results, irrespective of convenience and cost factors. Digital codes developed by other organizations, as part of a particular research study, were designed to satisfy different objectives and very often stressed ease of usage and economy of operation.

The main point to be made is that the state of the motor vehicle modeling art is not reflected in the various computer codes that have been generated, in that each code has been created to serve a particular purpose and thus each code has its own particular set of compromises. Some of the codes are in the public domain and others are proprietary, as, for example, the codes developed by the various motor vehicle companies. Some are well documented and some are not. Further, because it is not cost effective to write a generalized code which is capable of treating an arbitrary number of (1) vehicle elements and (2) wheels and axles, with any kind of suspension configuration and drive-wheel locations, codes are frequently specialized to handle particular configurations of vehicles. Finally, it should be noted that the simulation of commercial vehicles entails features of mechanical complexity that are not present in the four-wheeled motor car. Thus the state of the art

in simulating the commercial vehicle is constrained by our ability to measure, and/or define, the mechanical/kinematic properties of commercial suspensions and our ability to measure the properties of the large tires used on motor trucks. In summary, it may be stated that the state of the art available to predict the motion behavior of the motor vehicle is well developed from the theoretical and conceptual point of view but constrained, in a practical sense, by the need to (1) address specific mechanical configurations and (2) estimate, measure, or otherwise acquire, the data defining the vehicle-tire system under study.

No attempt shall be made here to define the state of the art in calculating, measuring, or estimating the various properties of the tire-vehicle system which appear in the mathematical description of this system. Rather, we shall identify, below, the various digital computer codes that are in the public domain and available to those who wish to conduct simulations without expending time and money to develop a code of their own. It is convenient to dichotomize this summary into "Passenger Car Simulations" and "Commercial Vehicle Simulations."

2.2.1 Passenger Car Simulations.

HVOSM - As mentioned earlier, this code has been developed at the Calspan Corporation to serve as a tool for analyzing pre-crash safety and post-crash performance after impacting certain kinds of fixed objects. In its present form [9], the program serves as a very comprehensive tool for predicting the braking and handling performance of the four-wheeled motor vehicle. The code contains a number of exclusive features such as terrain tables providing arbitrary roadway inputs and various tire modeling options to facilitate the computation of forces arising from (1) the traversal of an irregular roadway and (2) fore-aft/ lateral impact with curbs of arbitrary cross-section. The code also provides for user specification of a beam axle or an independent suspension at the front and rear of the car.

The shear forces at the tire-road interface are computed with the aid of an empirical tire model which was formulated at Calspan to fit the empirical data as generally supplied by another party, either fully

or in part. Although a spin degree of freedom is included for each wheel, the interaction between lateral and longitudinal slip is accounted for through the use of the "friction ellipse" concept. In this regard, the tire modeling can be viewed as not reflecting the latest state of the art. Nevertheless, the "friction ellipse" concept does appear to lead to results that agree reasonably well with measurement.

A unique feature of the HVOSM code is its graphic package which can convert trajectory computations into perspective views of the automobile with respect to the terrain and any objects that are included in the simulation. The documentation, after having been in a less than satisfactory state for many years, is now excellent as a result of the four-volume report issued by the Federal Highway Administration in February 1976 [9].

On the basis of the information available, it appears that this code can be used to determine the rollover threshold of four-wheeled motor vehicles with a reasonably high degree of confidence.

HSRI Passenger Car Simulation - The HSRI passenger car simulation has evolved from commercial vehicle simulations developed under sponsorship of the Motor Vehicle Manufacturers Association (MVMA). The program entails fifteen degrees of freedom including body motions, wheel jounce/rebound degrees of freedom, and wheel spin. Impact cannot be simulated and the range of validity of the roll angles is limited in that changes in the track due to roll, as seen in a plan view, are neglected. This simplification means that the numerical solution becomes invalid during the later stages of a rollover maneuver. Although the surface of the simulated roadway need not be smooth, no mechanism is provided for calculating the fore-aft and lateral forces caused by roadway undulations.

A semi-empirical tire model is used to calculate the shear forces at the tire-road interface across the entire range of longitudinal and lateral slip likely to be encountered in limit maneuvers. The resulting algorithm, which entails (1) user specification of the normal pressure distribution prevailing at the tire-road interface and (2) load-sensitive input parameters, is capable of matching measured tire data within five percent or less, constituting a substantial improvement over previously available algorithms. This added accuracy is extremely

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useful if the simulated maneuver covers all ranges of sideslip angles, rather than remaining entirely in a high- or low-angle range.

In comparison with HVOSM, this program is quite economical to run, which economy derives, in the main, from the methodology used to solve the wheel-spin equations in closed form, thus obviating the need for a very small time step as required to integrate the equations associated with wheel rotation. The documentation is complete and reasonably up to date. (See the appendices of Reference [10].)

This code will permit the user to obtain a good understanding of the maneuvers and dynamic conditions leading to rollover of the four-wheeled vehicle, characterized by an independent front suspension and a beam-axle rear suspension, but will not yield rollover thresholds, per se, because of simplifications that assume a limited angle of roll.

University of Tennessee Simulation Code - The title of this code derives from the academic affiliation of the developer of this program which was created at the National Highway Traffic Safety Administration (NHTSA). The model programmed into this code contains 19 degrees of freedom, including the usual ten degrees for the sprung and unsprung masses, plus time lags for the shear force build-up at each tire and a steering degree of freedom permitting the calculation of vehicle trajectory with the steering system unconstrained.

This simulation is distinguished by a very careful analysis of a large variety of front and rear suspensions, yielding equations of motion which are based on the assumption of an inclined roll axis. Tire shear forces are computed using the model employed in HVOSM.

Although no user-oriented documentation is known to exist other than an unpublished NHTSA report, a summary of the pertinent mathematical detail is given in Reference [11]. An examination of this reference indicates that no small angle assumptions are made. Thus, this code should, in principle, yield predictions of rollover thresholds. Comparisons with experimental measurements do not, however, inspire confidence in this regard.

NHTSA Hybrid Computer Simulation - (Whereas this listing of computer codes would ordinarily be restricted to digital programs which can be set up and run by any user, the hybrid-computer simulation mechanized at the Applied Physics Laboratory of Johns Hopkins University is a semi-permanent installation available to personnel under contract to the federal government and consequently this simulation does not require that any prospective user have a hybrid computer available for its operation.)

This simulation [12], in its current form, has evolved, over time, from a vehicle simulation which was originally developed by the Bendix Research Laboratories under contract support from NHTSA. The simulation is based on the mathematical model that was originally developed at Calspan for conversion into HVOSM. After Bendix modified the HVOSM program to fit on their hybrid-computer facility, NHTSA arranged for its transfer to APL in May 1972. Since that time the simulation has been updated by incorporating developments from the ongoing NHTSA-sponsored research program. At present, the following suspension types can be accommodated with this simulation:

- Independent suspension both front and rear
- Independent front suspension and solid rear axle
- Independent front suspension and solid rear axle with dual tires
- Solid axles, front and rear
- Solid axle, front, and solid axle, rear, with dual tires

Tire forces are modeled as per HVOSM.

On the basis of the information available, it appears that this computer simulation can be used to determine the rollover threshold of four-wheeled motor vehicles.

2.2.2 Commercial Vehicle Simulations.

AVDS-3 - The title of this code is an acronym for "Articulated Vehicle Dynamic Simulations" which were developed under NHTSA sponsorship at the Illinois Institute of Technology. The program has been written to simulate the dynamic response of combination commercial vehicles consisting of a truck-tractor towing one, two, or three trailers.

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Considerable simplifications are introduced (e.g., (1) no roll and pitch degress of freedom exist, requiring that the change in tire loads during maneuvers be computed on a quasi-static basis and (2) all units are restricted to have a single front and rear axle) to facilitate an inverse solution methodology. This solution methodology permits the user to specify the trajectory that he wants the vehicle to follow such that he can determine the steering and braking inputs which must be provided by the driver. In this manner, it is possible to determine whether a given vehicle combination makes unreasonable demands on driver abilities if a specific trajectory must be negotiated. This inverse procedure has been validated with generally good results [13].

The calculation of tire forces is based on a friction-ellipse representation of longitudinal and lateral force interaction, with no provision made for calculating longitudinal slip since a wheel rotation degree of freedom does not exist. The documentation for the latest version of this code was published in 1973 [14].

This code is not applicable to the prediction of a rollover threshold.

MVMA-HSRI Simulation - Under the auspices of the Motor Vehicle Manufacturers Association (MVMA), a project was undertaken at the Highway Safety Research Institute (HSRI) of The University of Michigan for the express purpose of establishing a digital computer-based mathematical method for predicting the longitudinal and directional response of trucks and tractor-trailers. Two computer codes have been produced, namely, a straight-line braking program for straight trucks, tractor-semitrailers, and tractor-semitrailer-full trailer combinations [15], and a combined braking and directional response program for trucks and tractor-semi-trailers [16]. The latter code is currently being augmented and updated under the auspices of the Federal Highway Administration to predict the braking and handling performance of the tractor-semitrailer-full trailer combination.

These codes have been designed to treat the various geometrical and mechanical features that are unique to the commercial vehicle. Among these features are the various proprietary tandem-axle suspensions (as commonly employed to perform a load-leveling function in the presence

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of roadway irregularities) with their unique nonlinear mechanical and
kinematic properties as depend on design particulars. Consequently,
each tandem suspension type or configuration must be individually
analyzed in order to model its mechanical and kinematic behavior. At
present, seven separate tandem suspensions may be selected as needed
to describe a given commercial truck or trailer, e.g., a "walking-beam"
suspension, a "four-spring" suspension, etc. Two tandem suspensions
are user options in the braking and directional response simulation—the
simplest four-spring suspension and a walking-beam suspension. Valida-
tion runs for these latter two suspensions have been performed in both
the straight-line braking and the braking and directional response
program.

The brake systems commonly employed on commercial vehicles require
special attention not usually necessary for vehicles with hydraulically-
actuated brake systems. The brake system model may conveniently be
divided into three sections. In a tractor-trailer air-brake system,
the driver applies the brakes by operating a treadle valve which con-
trols the air pressure at the brakes. In the first section of the model,
the relationship between pressure at the treadle valve and the line
pressure at the brakes on each axle is computed as a function of time.
The time delay and the rise-time characteristics of the air brake system
are represented in the simulation.

In the second section of the brake system model, the relation-
ship between line pressure and brake torque is modeled. The program
user has two options: he may either input a table of brake torque for
increasing line pressure, or ask the simulation to calculate a relation-
ship for torque versus line pressure, based on brake models contained
in the computer program.

The third section of the brake model contains the antilock brake
system simulation. This system is set up in a quite general form so
the user may call for any of a wide variety of antiskid control logic.
The documentation for these programs rests in several separate volumes.
The straight-line braking simulation code is described in Reference [15],
whereas the combined braking and handling simulation code is documented
in Reference [16].

Whereas this code permits the user to obtain a good understanding of the maneuvers that lead to a high probability of rollover for a truck and truck (tractor)-trailer combination, it cannot yield rollover thresholds, per se, because of simplifications that assume a limited angle of roll.

Cornell University Simulation - Work on the simulation of articulated vehicles has proceeded at Cornell University since the pioneering analysis and simulation of Mikulcik [17]. Several basic changes and refinements have been made, many of which have been discussed in the literature (e.g., [18], [19]).

The present code enables the user to "construct" the vehicle using a building block approach. Thus a straight truck, tractor-semitrailer, and doubles and triples combinations may be modeled with minimal inconvenience. It should be noted, however, that no provision is made for representing the peculiar properties of tandem suspensions and, in addition, all suspension springs and dashpots are assumed to have linear characteristics, an assumption that is in considerable conflict with reality. The axles and running gear can be assumed massless to reduce the degrees of freedom required to represent the total vehicle, which number can be sizeable since six degrees of freedom are used to describe each sprung mass of the total vehicle system.

The tire model is a modification of a formulation developed at HSRI [20]. Since the formulation makes use of a closed-form integration of the shear stresses at the tire-road interface, no "friction ellipse" type of calculations are necessary to compute the interactions between the lateral and longitudinal forces.

An explanation of this model is presented in Reference [21]. There is, however, no published information in the form of a user's manual.

In addition to the inability of this code to describe commercial vehicles in a realistic manner, it is believed that the assumptions made in describing the properties of the fifth-wheel coupling are likely to invalidate the prediction of a rollover event.

STI Simulation - An analysis of truck and bus handling was performed by Systems Technology, Inc. (STI) under NHTSA sponsorship. In the course of this contract, nonlinear equations of motion were derived for a three-axle straight truck and an intercity bus and implemented as digital computer simulations.

Tire shear forces are computed based on the Calspan tire model as presented in Reference [9]. This code is distinguished by its capability to compute the effects of sloshing of liquid cargo as explained in Reference [22]. In particular, the liquid cargo is assumed to be inviscid and incompressible and the fluid flow is assumed to be irrotational. A cylindrical tank with a circular cross section is assumed and equations are initially derived for an arbitrary liquid cargo level and then specialized to the half-full case. The angular displacement of the plane of the water surface defines the wave motion, and these pitch and roll angles are assumed to be small perturbations. Only the first harmonics of these wave motions are included in the analytical representation.

The STI code appears to yield good correlation with full-scale test results. A source list of the program, and other pertinent details, are presented in Reference [22].

The documentation of this code indicates that it should be able to predict the rollover threshold of three-axle vehicles.

Roll and Yaw-Plane Analysis: Multi-Element Articulated Vehicle - A study performed by HSRI recently under the auspices of the State of Michigan has resulted in two computer codes which can be used, in combination, to evaluate the likelihood of rollover of a multi-articulated vehicle. The first code, as implemented, yields the directional response to steering as predicted by the linearized equations of motion applicable to a tractor-semitrailer-dolly-semitrailer combination with an arbitrary number of axles. The second code yields the lateral, bounce (or heave), and roll motions of a suspended mass as caused by side forces acting at the tire-road contact which are equivalent to the lateral acceleration time histories produced by the linear, planar analysis. This latter code stresses the nonlinear character (e.g., suspension lash, dry

friction, and suspension rate hardening) of actual truck suspensions and the lateral, vertical, and roll degrees of freedom of the sprung and unsprung masses, both of which are free to roll relative to ground. Clearly, the development of these two codes, one being based on a linear analysis, was seen as a highly pragmatic way to arrive at roll-over predictions without necessitating the development of a set of non-linear equations describing a multi-articulated vehicle system having, at minimum, twenty-five degrees of freedom.

These two codes have been shown to yield predictions of lateral response and roll response, respectively, of a "pup" trailer that are in reasonably good agreement with experiment. The codes are described in Appendices A and B of Reference [23].

It appears that these two codes, in combination, provide considerable insight with respect to the manner in which yaw response characteristics and design details (related to rolling behavior) combine to produce a high probability of rollover. Specific rollover thresholds cannot be established with a high degree of accuracy, however.

2.3   Properties of the Motor Vehicle Influencing the Rollover Threshold

Other than the last mentioned item in the listing of computer codes given above, none of the abovementioned simulations were developed for the express purpose of examining the rollover threshold of the motor car (or truck). It can also be said that, to the degree that some of the above codes are valid for conducting such an examination, it does not appear that a complete systematic study has ever been made to determine the sensitivity of rollover thresholds to design variables.

Notwithstanding the absence of such a study, a certain amount of knowledge and experience has been obtained to indicate the general nature of the rollover process and to show that rollover can (and does) occur dynamically, even though steady turning at the limiting lateral acceleration may be insufficient to produce a rollover response. Roll-over under dynamic conditions is also known to be a function, in part, of the dynamic maneuver that is performed. In other words, the rollover threshold is maneuver sensitive and, consequently, some care must be taken in defining the rollover threshold of a motor vehicle.

As indicated earlier, research has been performed to define the combined steering and braking maneuver which would appear to make maximum demands on a motor car's natural immunity to rollover. Simulation of the directional response produced by steering and braking inputs sufficient to create a rollover, as determined experimentally, has produced computer output that agrees rather well with measured response motions (see Appendix B of Reference [10]). It must be noted, however, that these simulations were accompanied by a sizeable parameter data gathering effort which included, among other measurement activities, a substantial effort to measure the nonlinear mechanical properties of the appropriate pneumatic tire on the actual surface used in the test program.

Given that it has been demonstrated that the availability of valid parameter data enables one to predict the "drastic steer and brake" roll response of a specific category of motor car, namely, a motor car with independent front suspension and a beam-axle rear suspension, it is reasonable to conclude that this prediction can also be made for a vehicle with an independent rear suspension, provided the kinematic properties of such a suspension are properly accounted for in the applicable equations of motion. Even though this exercise appears to have never been carried out, it is possible to enumerate the properties of a motor vehicle which, to first order, determine its roll behavior and, more importantly, its rollover threshold in a given maneuver.

To the degree that the physics of the motor vehicle are reasonably well understood at this point in time, the following properties of the motor vehicle are primary to the establishment of its immunity to (or, conversely, its proclivity for) rollover. In order of probable decreasing significance, they are:

- center of gravity height and track width

- magnitude of the frictional coupling between the installed tires and the road surface

- geometry of the front and rear suspensions as establish their respective roll centers and the track and camber change resulting from jounce/rebound motions of the front and rear wheels

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- the stiffness and damping properties of the front and rear suspensions, as establish the roll stiffness and roll damping of the front and rear suspensions, respectively, over the full range of jounce/rebound displacement

- the limits on jounce/rebound displacement of the front and rear wheels, including the additional stiffness created by contact of the jounce/rebound stops (i.e., bump stop contact)

- the anti-pitch properties of the front and rear suspensions

- the roll inertia of the sprung mass plus the roll inertia of unsprung assemblies, such as beam axle with wheel and tire masses.

In addition to the above-cited properties, one must also measure or estimate those properties which are essential to predicting the overall directional response of the motor vehicle. Chief among these properties are the lateral mechanical properties of the installed tires, the yaw inertia of the sprung and unsprung masses, the geometry of the vehicle in plan, and the suspension geometry that influences wheel and axle motions, as viewed in the plane of the roadway.

The state of the art in measuring and/or calculating these properties is reasonably well developed in principle, but, in practice, it appears that only a limited number of organizations have invested the time and money necessary to create the laboratory facilities needed to make these measurements. The reason for this state of affairs derives from the fact that the primary user of such facilities is, logically, the motor vehicle manufacturing industry. To the extent that this industry sees simulation activities and/or the gathering of vehicle parameter data as important to the vehicle development process, it does invest in the establishment of the necessary measurement facilities. By and large, however, it is so much simpler and easier to conduct performance tests with either prototypes or final products than it is to predict performance with the aid of a computer simulation, that many companies do not spend the time and money necessary to make parameter measurements and, in some cases, do not spend the time and money to create this particular capability. In a few instances, research

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organizations and academic institutions have seen fit to establish a
portion of the required measurement capability and the advent of federal
safety standards has led to an increased interest in (1) advancing
the real-world practice of parameter measurements and (2) creating more
of the required facilities. Notwithstanding these trends, it can be
stated that, for many purposes, it is much more cost effective to
measure the rollover threshold of a motor vehicle than it is to predict
this threshold by means of a computer simulation. On the other hand,
an indepth understanding of why the vehicle behaves as it does can,
clearly, be better obtained from a simulation endeavor.

3.0 SIMULATION OF THE M-151

The M-151 is the military vehicle selected for scrutiny in this study in light of its operational history which appears to contain a larger than expected number of rollover incidents. Given that the M-151 is a four-wheeled vehicle with an independent front and rear suspension, the HVOSM (identified in Section 2.0 above) is particularly applicable to simulating this vehicle. Accordingly, we discuss below certain items relating to using this code in pursuing the objectives of this study.

3.1   The HVOSM Code

The Highway-Vehicle-Object Simulation Model exists in two forms—a road design version (in which vehicle impact with roadside objects is included) and a vehicle dynamics version, HVOSM VD2. A digital magnetic tape containing the source code for both versions was obtained from the Federal Highway Administration (FHWA), together with four volumes of program documentation. This tape also contained the source code for a preprocessing program which can be employed to (a) retrieve vehicle design data relating to each of six cars which have been extensively measured or (b) construct typical automobile parameters corresponding to a specified wheelbase. To facilitate this study, the source codes for HVOSM VD2 and the preprocessing program were copied to disc files on the Michigan Terminal System (MTS), and the tape was returned to FHWA.

Some small modifications to the HVOSM VD2 (concerning the printing of the date and the writing of results to the line printer) were necessary before the program would compile and run. The standard form of output generated by the program consists of successive values of the most significant variables describing the motions of and forces on the vehicle as written by the line printer. Appropriate headings are also printed, and the user has some facility for suppressing unwanted output. The line printer output can be viewed at the user's terminal immediately after a simulation run, but since the output format is designed to use all of the line printer's 120 columns, the terminal output is not very

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convenient, and can only normally be used to determine whether or not rollover occurred. However, by post-processing the results, and employing HSRI's graph plotting routine, computer-plotted time histories can be obtained. Although the time histories which are included in this report have been obtained in this way, the output of the line printer constituted the main interface between the computer and the analyst during the course of this project.

3.2   Acquisition of Input Data Defining the M-151

To obtain values of the parameters describing the M-151, reference has been made to the following documents: U.S. Army Test and Evaluation Command Report No. DPS-2642 [24]; Stevens Institute of Technology, Davidson Laboratory Report 1420 [25]; Highway Safety Research Institute Report UM-HSRI-PF-74-3 [26]; and the M-151 Operator's Manual [27].

Reference [24] contained information on vehicle weight and c.g. position, suspension spring rates, shock absorber force/velocity characteristics, and jounce and rebound buffer locations. Reference [25] contained limited information relating to a linear analysis of the M-151, which information was clearly in error, in many instances. For example, the total mass, as given, is much lower than that given by Aberdeen, the wheelbase quoted is five inches too small, the roll inertia is over 90% of the yaw inertia (which is plainly absurd) and certain of the suspension data are clearly incorrect. Consequently, little reliance has been placed on this information. Reference [26] contains tire forces and moments measured on free-rolling 7.00 x 16 NDCC tires at 20 and 25 psi pressure (these are the standard tires and inflation pressures for the M-151) for a limited range of loads, sideslip angles, and camber angles. Reference [27] contains sketches and diagrams which have been of some use in the estimation of those vehicle parameters for which no measurements are known to exist.

In addition, informal arrangements were made with the Reserve Training Center of the Department of the Army (in Ann Arbor) to take measurements and photographs of an M-151, particularly the suspension

geometry, such that suspension layouts could be drawn and the necessary data derived from the drawings. These measurements and photographs were also useful for estimating inertia properties. Subsequently, these preliminary estimates were improved on the basis of measurements (performed here at HSRI in connection with another project) of the pitch inertia of an AMC Jeep.

To satisfy the input requirements of the HVOSM code, a considerable amount of data—non-critical as far as the project objectives were concerned—had to be constructed. These data related to the properties of the transmission system, brake system properties, engine torque characteristics with open and closed throttle, and vehicle drag and rolling resistance. Estimated values for these properties are considered reasonable, rather than accurate.

With respect to the parameters that affect steering responses and rollover rather markedly, however, a few problems were encountered. For example, the lateral force properties of the M-151 tire, as measured, were found not to fit the HVOSM code particularly well in respect of the variation of cornering stiffness with loading and camber effects. Further, early computer runs indicated, in the extreme maneuvers being considered, that (1) the tires were operating at much higher loads and slip angles than we had experimental data for and (2) the manner in which the tire input data was being extrapolated (within the program) was not reasonable. Through reference to other tire data defining side forces generated at high loads and slip angles (particularly Reference [28]), the tire data was modified so that the program was required, for the most part, to interpolate rather than extrapolate. It should be borne in mind, therefore, that the tire data which is critical to accurate prediction of vehicle motions, is, to some extent, estimated. Somewhat less critical, but, nevertheless, important, estimates had to be made of suspension jounce- and rebound-stop stiffness and resiliences in a somewhat blind manner. In addition, a value for shock absorber damping must be selected as a compromise between the damping existing during jounce and rebound motions since HVOSM does not allow asymmetry in the behavior of the shock absorber. Suspension friction, which will vary somewhat from one jeep to another, also had to be estimated. Throughout the study, the road surface has been assumed flat and dry.

4.0 MANEUVERING CHARACTERISTICS OF THE M-151

Four different maneuvers were simulated with the M-151 in order to explore its potential for rolling over or resisting rollover. The motion behavior of the M-151 as discerned in

1) a combined braking and steering maneuver,

2) a quasi-static turning condition,

3) a J-turn maneuver, and

4) a nominal lane-change maneuver

is discussed below by treating each of the above maneuvers in the order listed.

4.1 Combined Steering and Braking Maneuver

At the outset, it was anticipated that a combined steering and braking maneuver would present the greatest challenge to the roll stability of the M-151. This expectation was mainly based on the experimental results reported in References [1] and [2]. The maneuver starts from a straight path and consists of throttling back and applying a half sine-wave steering input and then applying the brakes for 1/2 sec. sufficiently hard to lock all the wheels starting when the vehicle has reached its maximum yaw rate in response to the steering input. For a given vehicle speed, the amplitude and period of the sine wave, and the timing of brake application and release will all affect the responses in a complex fashion, since the phasing of motions and applied is critical to determining whether disturbances are additive or cancel. Even for one vehicle in one loading condition, it would, in general, require a large number of open-loop control computer runs for each vehicle speed to establish "optimum" conditions for rollover. This approach has not been attempted; rather, a small sample of results have been obtained for the standard M-151 at different speeds disturbed by the same steering and braking sequence.

For an initial speed of 50 mph, the sequence leads to rollover. The motion time histories are shown in Figure 2. In general terms, the event sequence can be described as follows. The steering input
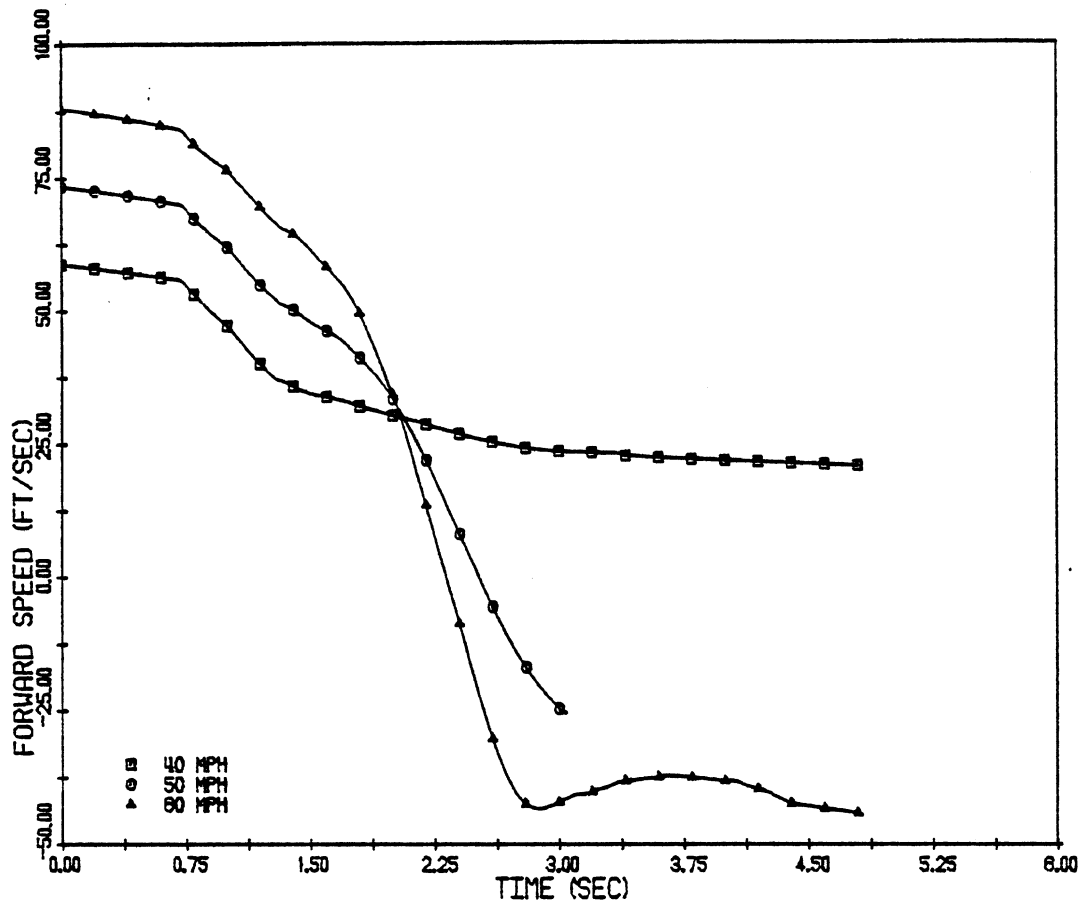
M151 SIMULATION, COMBINED STEER AND BRAKE, 40, 50, AND 60 MPH

Figure 2(a)
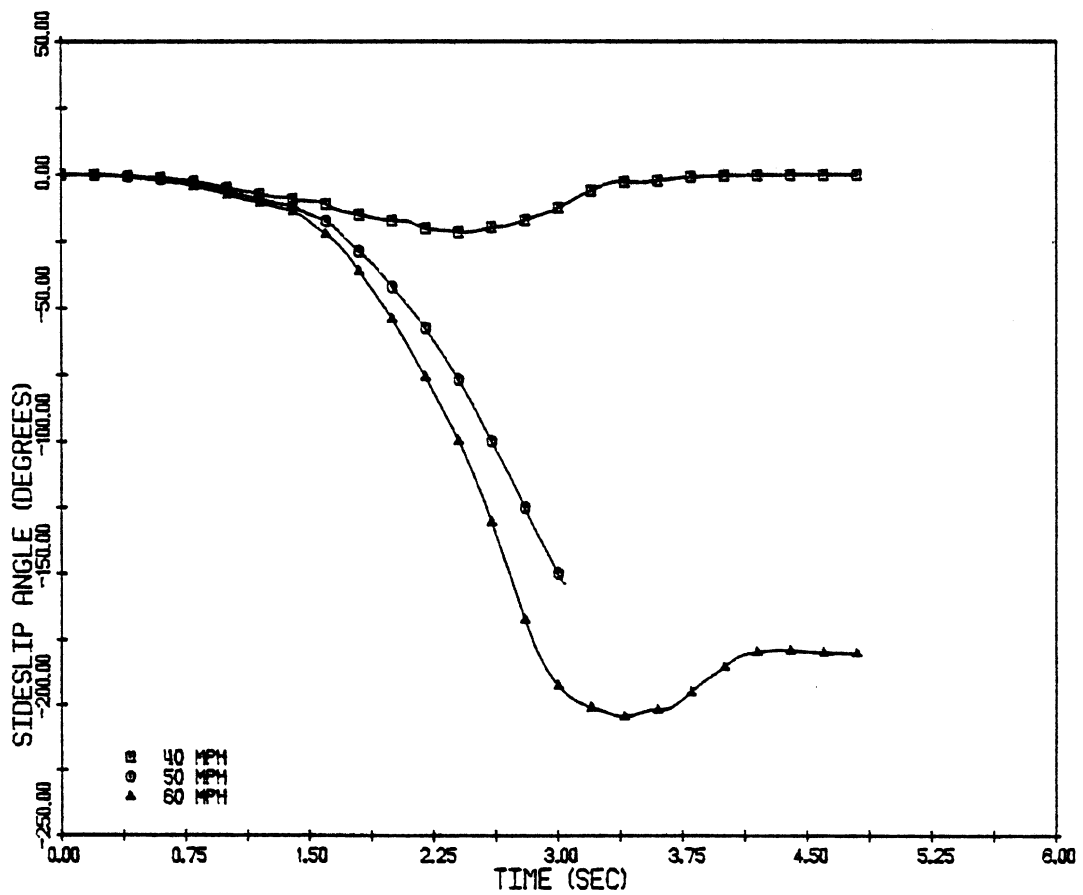
M151 SIMULATION, COMBINED STEER AND BRAKE, 40, 50, AND 60 MPH
Figure 2(b)
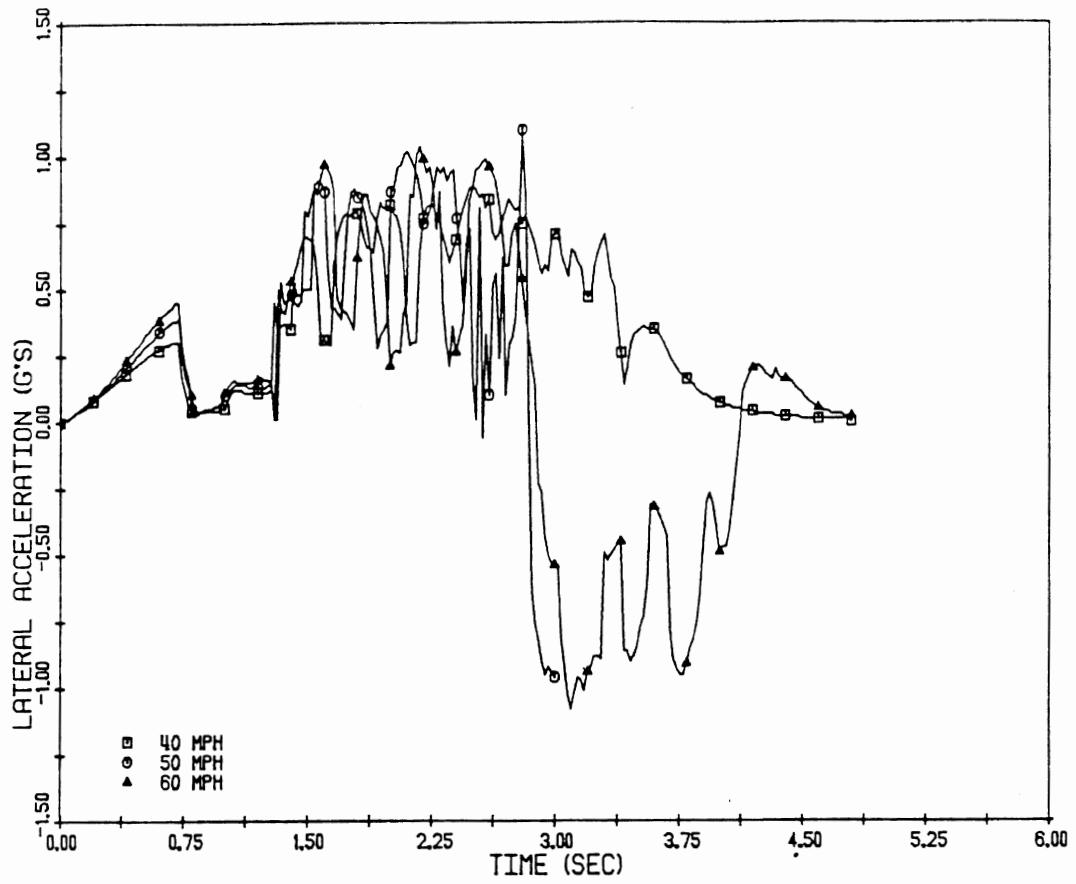
M151 SIMULATION, COMBINED STEER AND BRAKE, 40, 50, 60 MPH

Figure 2(c)
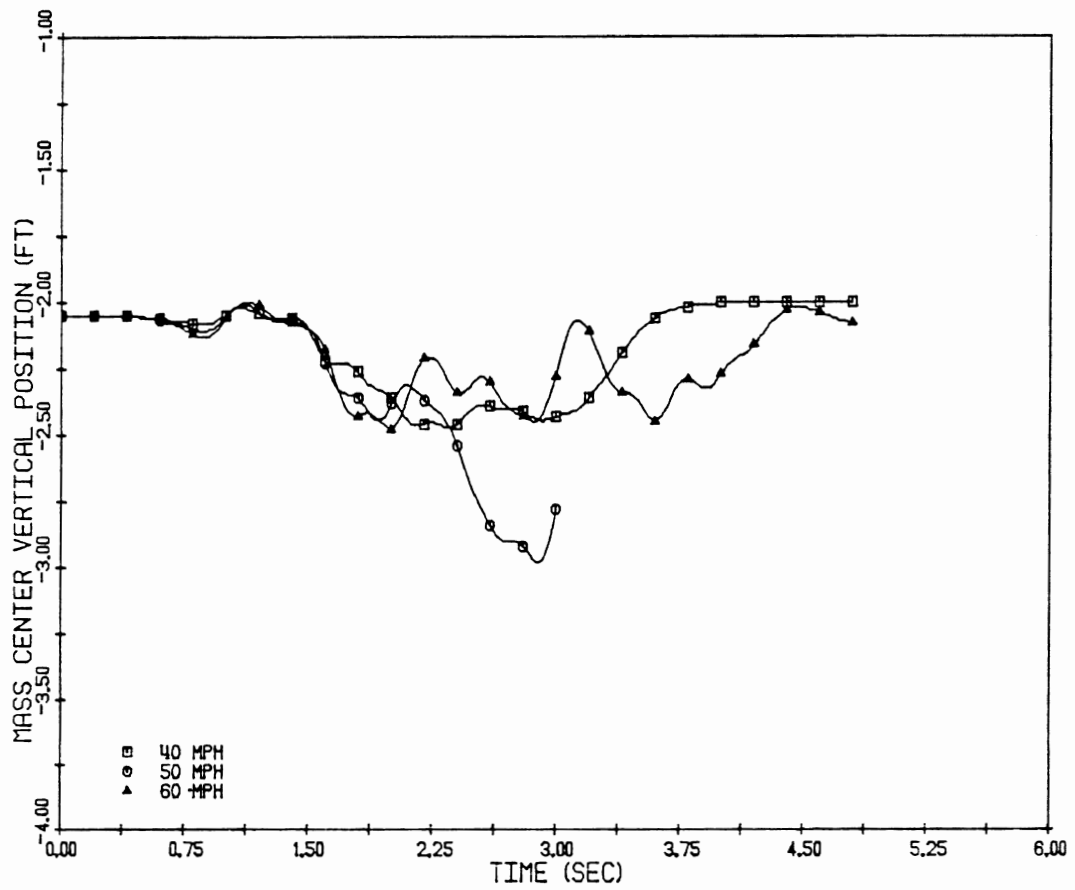
M151 SIMULATION, COMBINED STEER AND BRAKE, 40, 50, 60 MPH

Figure 2(d)

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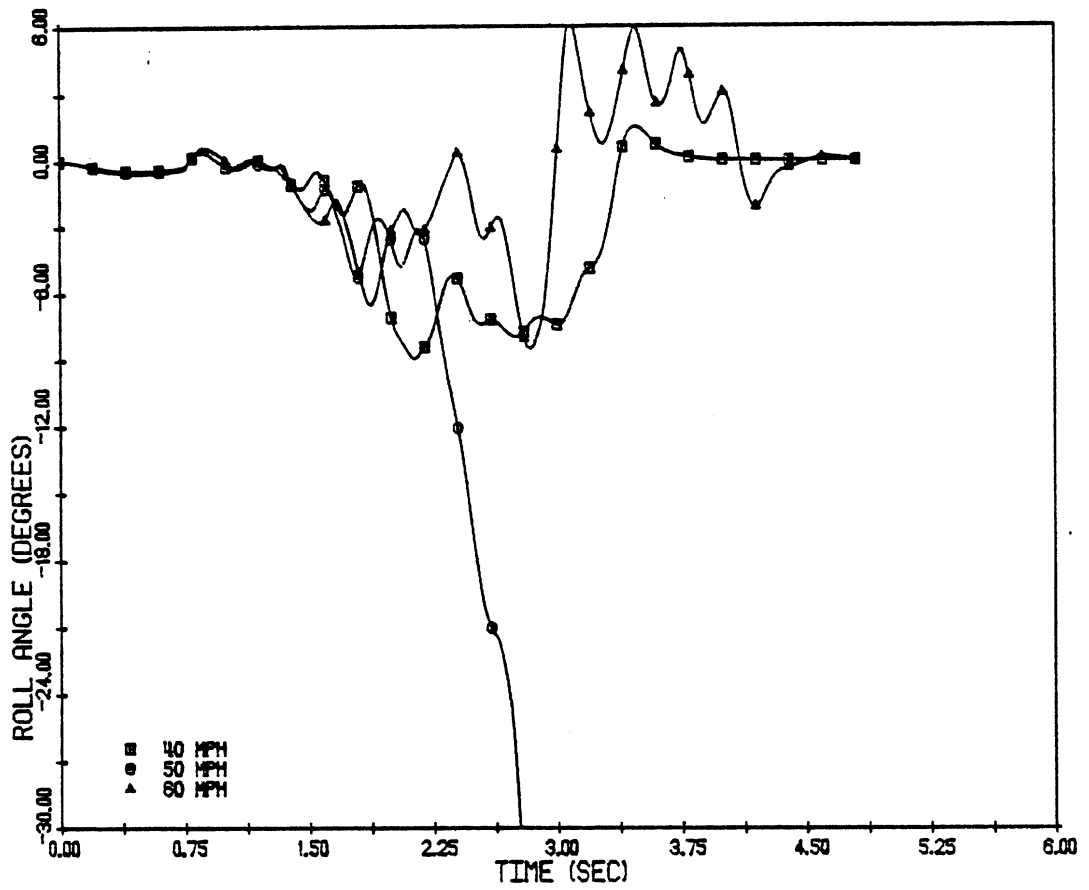
M151 SIMULATION, COMBINED STEER AND BRAKE, 40, 50, 60 MPH

Figure 2(e)
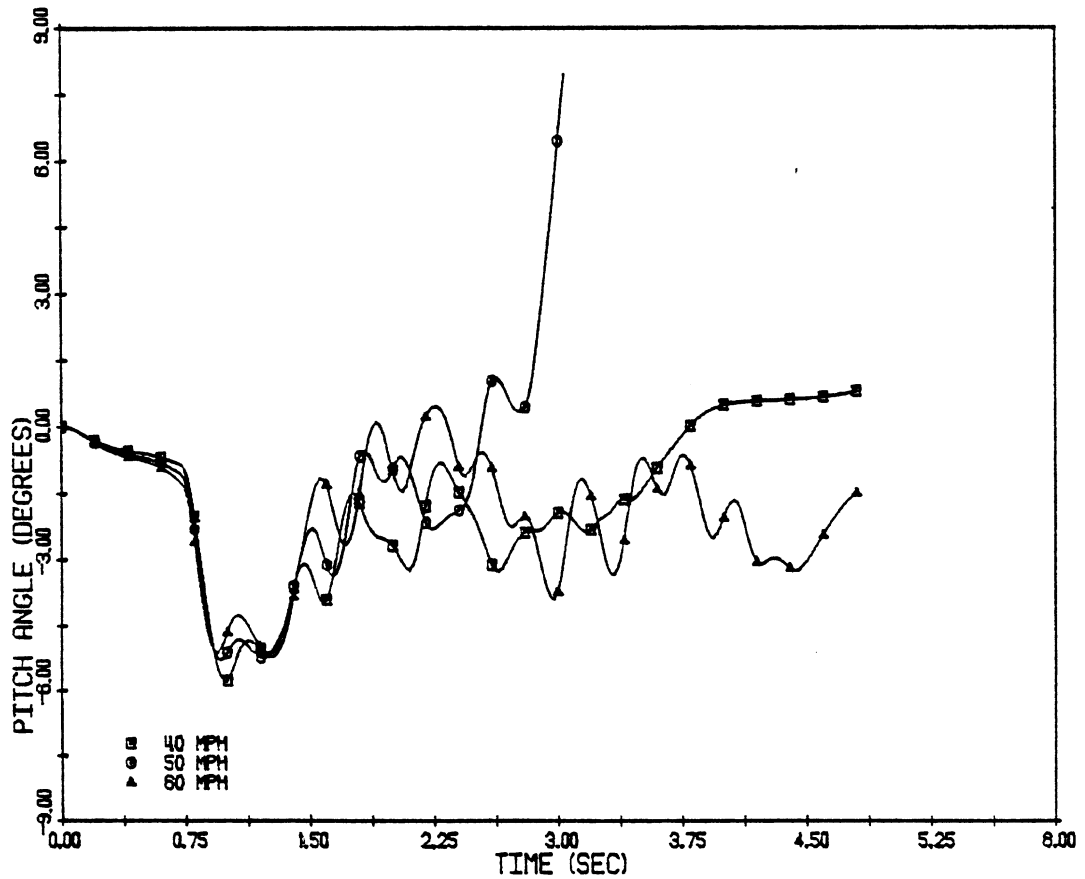
M151 SIMULATION, COMBINED STEER AND BRAKE, 40, 50, 60 MPH

Figure 2(f)

causes the vehicle to yaw without very much attitude change. When the wheels are locked by the heavy braking, the tires produce very little side force and the yaw rate remains fairly constant, while the vehicle pitches nose down in response to the braking forces. When the brakes are released, the front wheels spin up to somewhere near their free-rolling velocity more quickly than the rear wheels because they are more heavily laden, and have less spin inertia through not being geared to the transmission system. The front side forces, therefore, grow before the rear forces and act to increase the yaw velocity. Also, when the brakes are released, the magnitude of the pitch angle decreases in a somewhat oscillatory fashion. The tire side forces then sustain a nose-down pitch attitude, with the vehicle center of mass raised about four inches from the equilibrium state, as a result of the strong jacking effect occurring at the rear suspension. The roll angle builds up, again with some oscillation, until rollover occurs.

At 40 mph, the event sequence was qualitatively similar during the initial protion of the response. The main quantitative difference was that the final build-up of roll angle was not sufficient to cause rollover, although at one point three of the four wheels lost contact with the ground.

When the starting speed was 60 mph, a similar sequence occurred until a little after the brakes had been released. Then the vehicle turned through a greater angle relative to its path and traveled backwards (having spun). Although high lateral accelerations were reached many times, they were not sustained, and not phased with the rolling motions of the sprung mass such that rollover occurred. There is little doubt that rollover could be achieved from 60 mph by employing different magnitudes and timings in the control inputs.

4.2 Steady Turn Behavior

Steady turning does not create the greatest likelihood of roll-over, but, in an open-loop control simulation study, it has the advantage of being the easiest maneuver to describe quantitatively, because the very important phasing of time-varying effects, already mentioned, is absent in this case. For reasons of economy, the

steady-turn behavior of vehicles is often studied experimentally, employing quasi-steady conditions. Two forms of the quasi-steady turn test exist: (1) maintaining constant speed with slowly increasing steering-wheel displacement, or (2) maintaining constant steering-wheel displacement with slowly increasing speed. In such quasi-steady tests, rates of change must be maintained small enough for time-varying effects to be negligible.

Simulations of the M-151 were carried out first by employing fixed steering and open throttle, the intention being to increase the forward speed until rollover, spinout, or any other limiting response was reached. It was found that an initial equilibrium of the vehicle could only be established for lateral accelerations less than about 0.6 g, in which condition the front and rear tire sideslip angles were roughly 6.5° and 10°, respectively. The tires generate their maximum side forces at much greater slip angles, of course, but if the lateral acceleration is any greater than 0.6 g (quasi-steady), the inside rear tire lifts off the road and the engine torque simply spins the inside rear wheel. In the real world, a driver would throttle back to prevent the engine from overspeeding and the vehicle would slow down. The lateral acceleration would reduce again to a level at which the inside rear wheel would regain road contact and would transmit some drive thrust. Thus a region of interest involving a rollover could not be reached in a maneuver consisting of a fixed steer angle and slowly increasing throttle.

The alternative form of quasi-steady turning in which a constant speed is maintained, while steer angle is slowly increased, was also tried, but the inside rear wheel again lifted and prevented sufficient engine power from reaching the road to maintain speed. After wheel lift-off occurs, the slowly increasing steer angle causes the speed to decrease such that a steady lateral acceleration could be achieved and maintained, but not increased. If the steer angle were to be increased sufficiently quickly to cause an increasing lateral acceleration, and some limiting response, the maneuver ceased to be a quasi-static maneuver.

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In circumstances in which the steer angle increases at such a rate as to maintain a constant lateral acceleration of 0.72 g, by virtue of a steadily decreasing forward speed, the external force system on the vehicle changes very little with time. The vehicle attitude is as near to an equilibrium attitude (corresponding to that lateral acceleration) as can be established. In this situation, the mass center of the body is raised 3.2 inches from its static equilibrium position, and the body is pitched nose-down through 2.3°. The roll angle is 5.6° and the wheels have camber angles of 6.2° (left front), 3.9° (right front), 19.2° (left rear), and -6.5° (right rear)*· and normal loads of 1580 lb_f, 190 lb_f, 1635 lb_f and 0 lb_f, respectively.

These quasi-static turning maneuvers also show that, for lateral accelerations above about 0.55 g, the "equilibrium" roll angle changes rapidly with lateral acceleration from about 1° up to about 6° at the cornering limit. Clearly, up to 0.5 g of lateral acceleration, very little body roll occurs. This property can be expected, in practice, to make the M-151 very difficult to control in limiting conditions, because its attitude, and therefore its response behavior, can vary through chance circumstances relating to wind, road surface, previous motion, etc., for different approaches to a particular maneuver.

4.3   The J-Turn Maneuver

For reasons of economy, a J-turn maneuver has been simulated by assuming the M-151 to be traveling straight and level at constant speed and at time zero (the start of the simulation run) having the throttle closed, and a step input of steer angle applied at the road wheels. In practice, a step input is not possible, but a realistic fast ramp input of steer angle produces substantially the same results as a step input, since the input would be completed before much vehicle response had occurred.

A J-turn maneuver for a particular vehicle configuration is characterized by the vehicle's initial forward speed and the magnitude of the steering input. Thus, it is simple to describe. Also, in the case of the M-151, quite general motions of the vehicle, including

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*The minus sign indicates that the wheel plane is inclined in the direction opposite to that of the body roll.

jounce and pitch of the body, could be excited due to the strong coupling of longitudinal and lateral motions resulting from the suspension kinematics. This maneuver has therefore been studied for a range of forward speeds and steer angles.

For relatively small steering angles, which lead to responses in which the tire sideslip angles never exceed six degrees and the lateral acceleration does not exceed 0.5 g, the forward speed decreases steadily while the lateral velocity and lateral acceleration rise steadily. The vehicle pitches nose down and hardly rolls at all. The tire sideslip angles are not far from equal at the front and rear, although as the lateral acceleration rises from 0.2 g, there is a noticeable tendency for the rear tires to sideslip more than the front tires. Lateral load transfer at the rear of the vehicle greatly exceeds that at the front.

For larger steering inputs, which lead to rollover or near rollover responses, the behavior is qualitatively the same irrespective of the vehicle speed or the steer angle applied. In some quantitative respects (e.g., lateral acceleration, pitch angle, roll angle, jacking, suspension behavior, load transfer), the behavior is substantially constant, but the time taken for the limiting condition to be reached varies with speed and steer. Typical rollover and less severe responses are shown in Figure 3. The high level responses can be described as follows. A one-degree nose down pitch attitude develops rapidly, steadies off, and then builds up further to about 2.7 degrees, as the lateral acceleration builds up steadily to 0.75 g. The roll angle remains very small until the lateral acceleration reaches about 0.45 g, and then increases rapidly to rollover if the maneuver is severe enough. At about 0.65 g, the inside rear wheel leaves the ground, and a short time later, the inside front wheel also lifts. Just prior to rollover, the pitch angle changes sign, and the vehicle rolls with a nose up attitude.

For each initial speed above some minimum value, there is a critical steer angle for the J-turn maneuver below which the vehicle will not rollover and above which it will. By trial and error, the critical input has been determined for the standard M-151 for speeds between
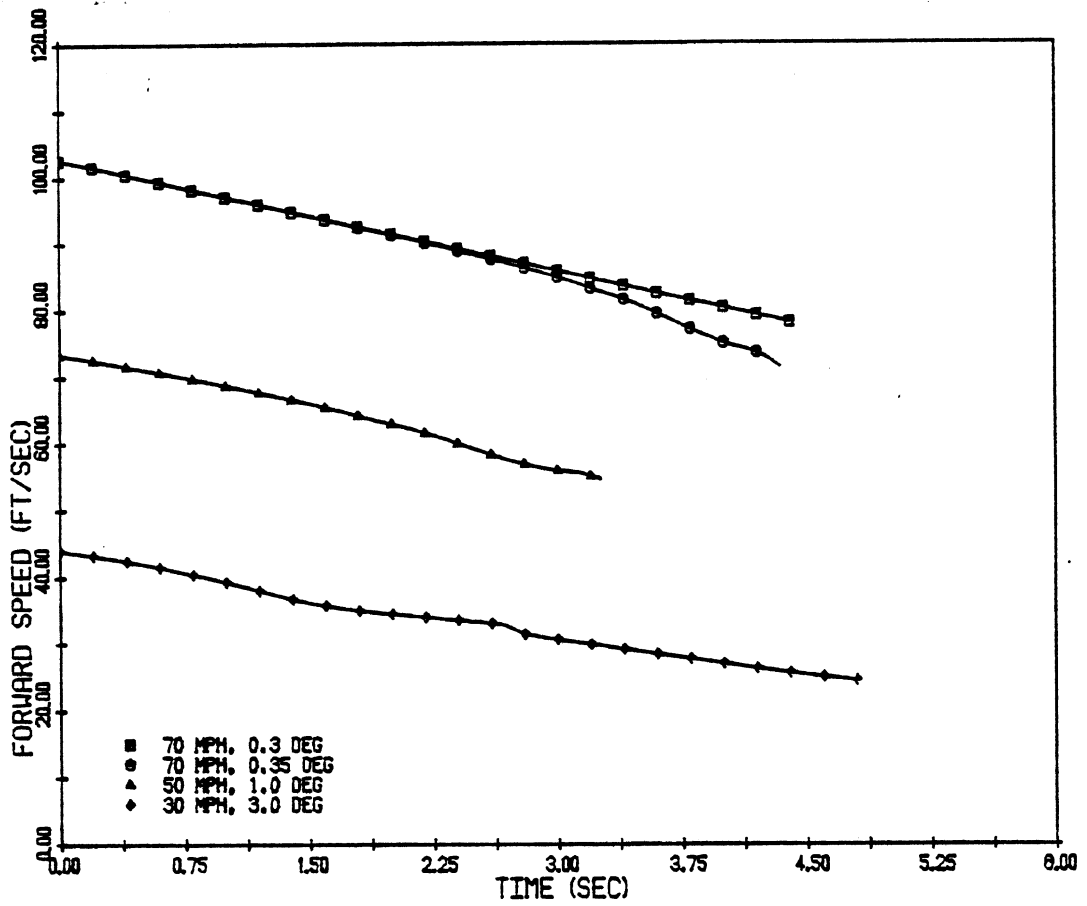
M151 SIMULATION, J-TURN AT 30, 50, AND 70 MPH

Figure 3(a)
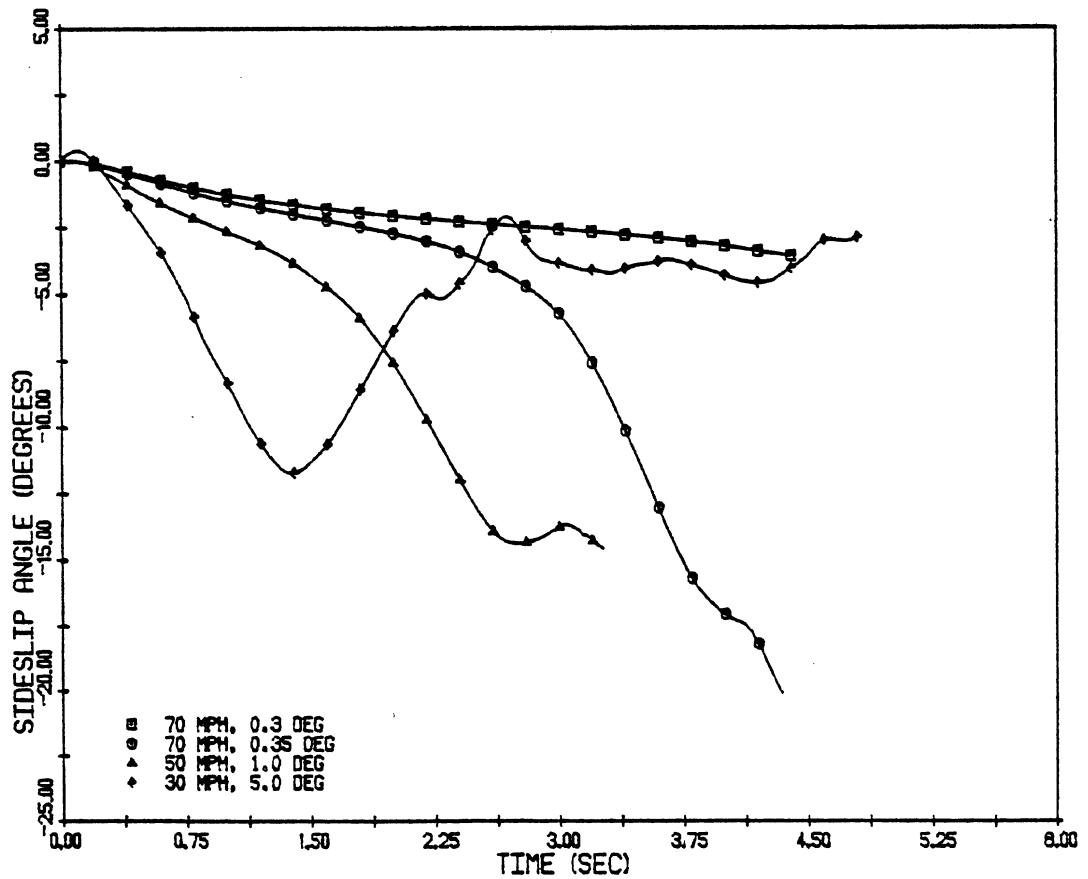
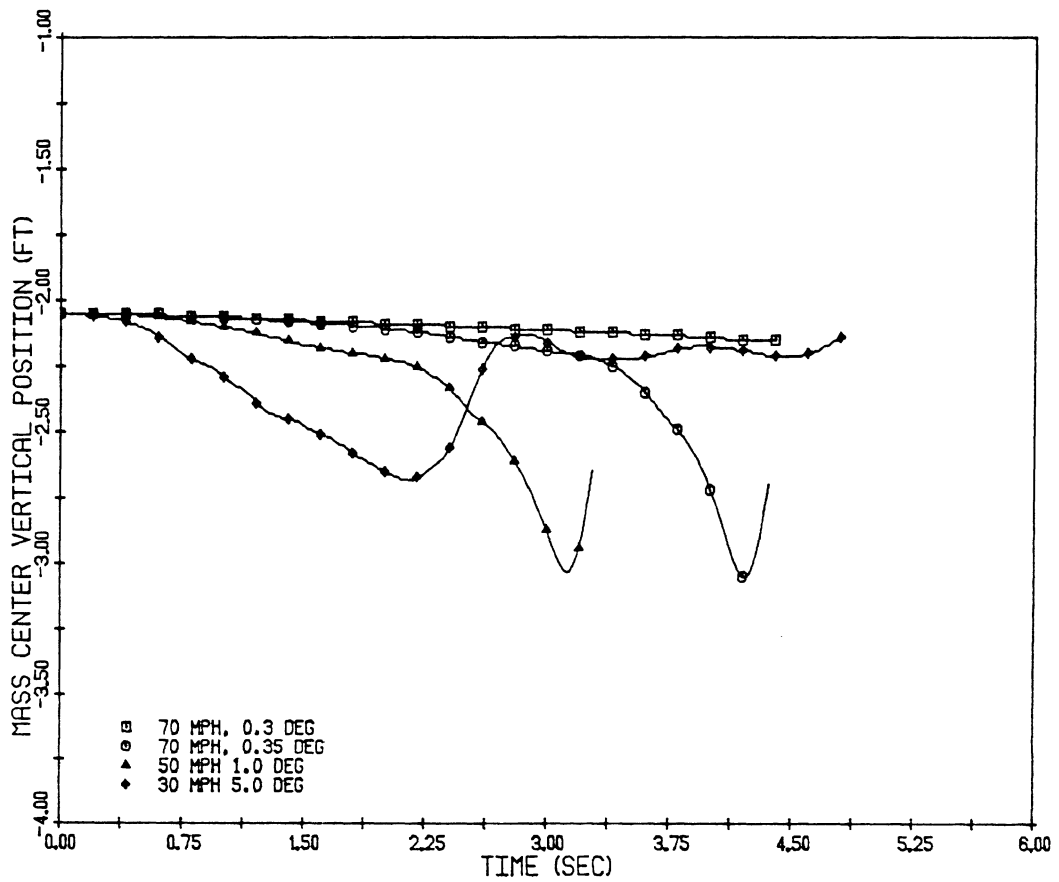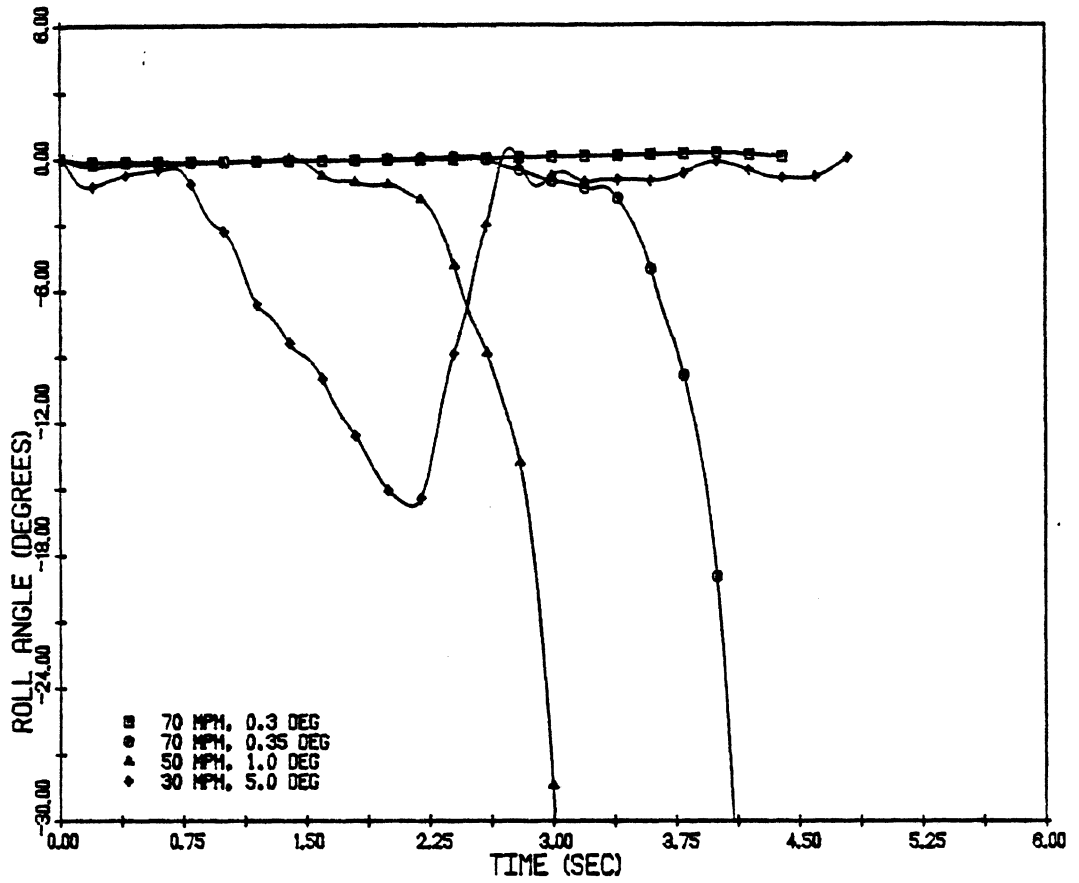M151 SIMULATION, J-TURN AT 30, 50, AND 70 MPH

Figure 3(b)

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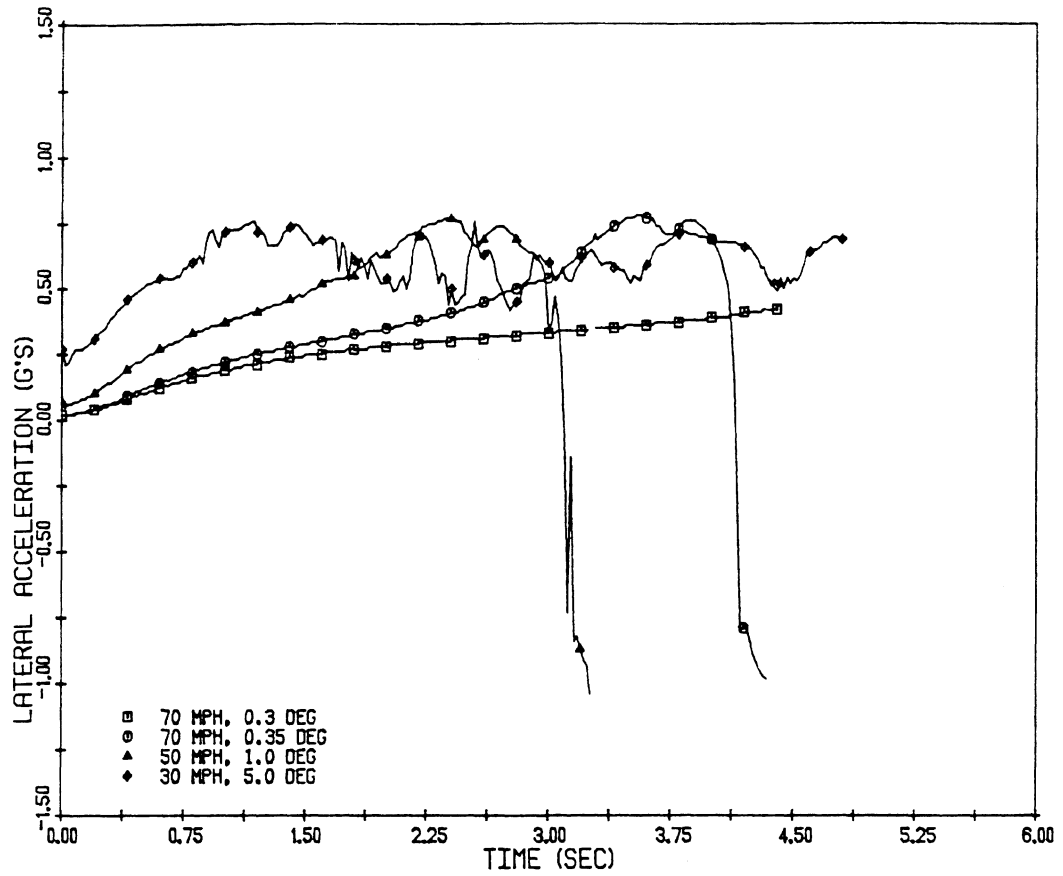
M151 SIMULATION, J-TURN AT 30, 50, AND 70 MPH
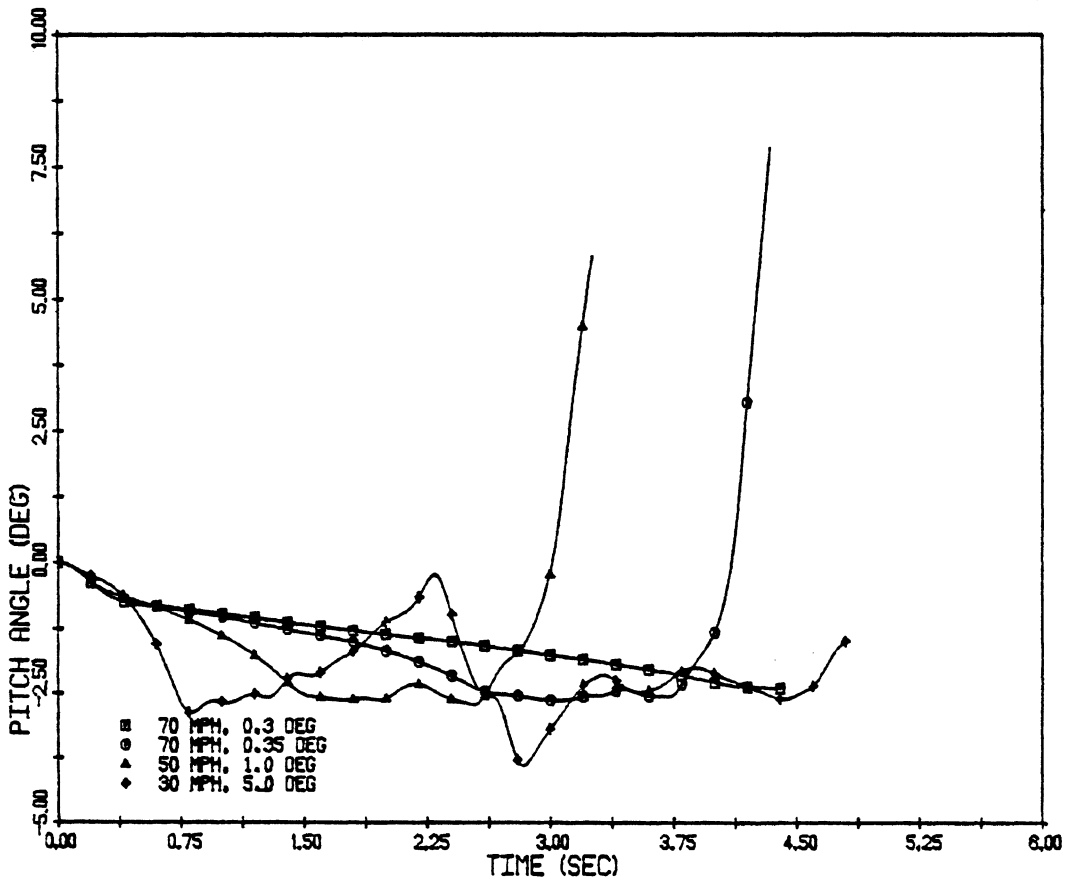
Figure 3(c)

M151 SIMULATION, J-TURN AT 30, 50, AND 70 MPH

Figure 3(d)

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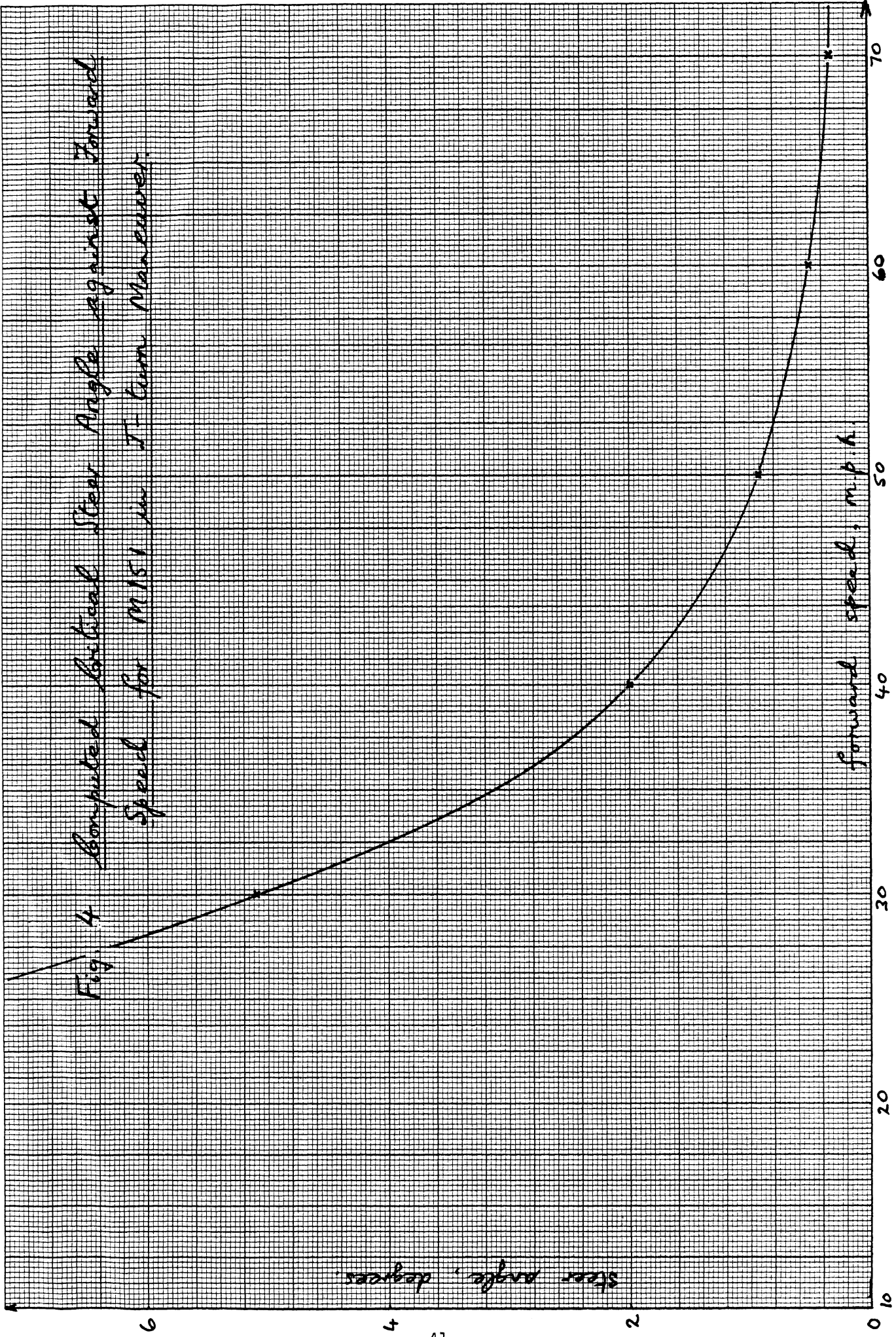M151 SIMULATION, J-TURN AT 30, 50, AND 70 MPH

Figure 3(e)

M151 SIMULATION, J-TURN AT 30, 50, AND 70 MPH
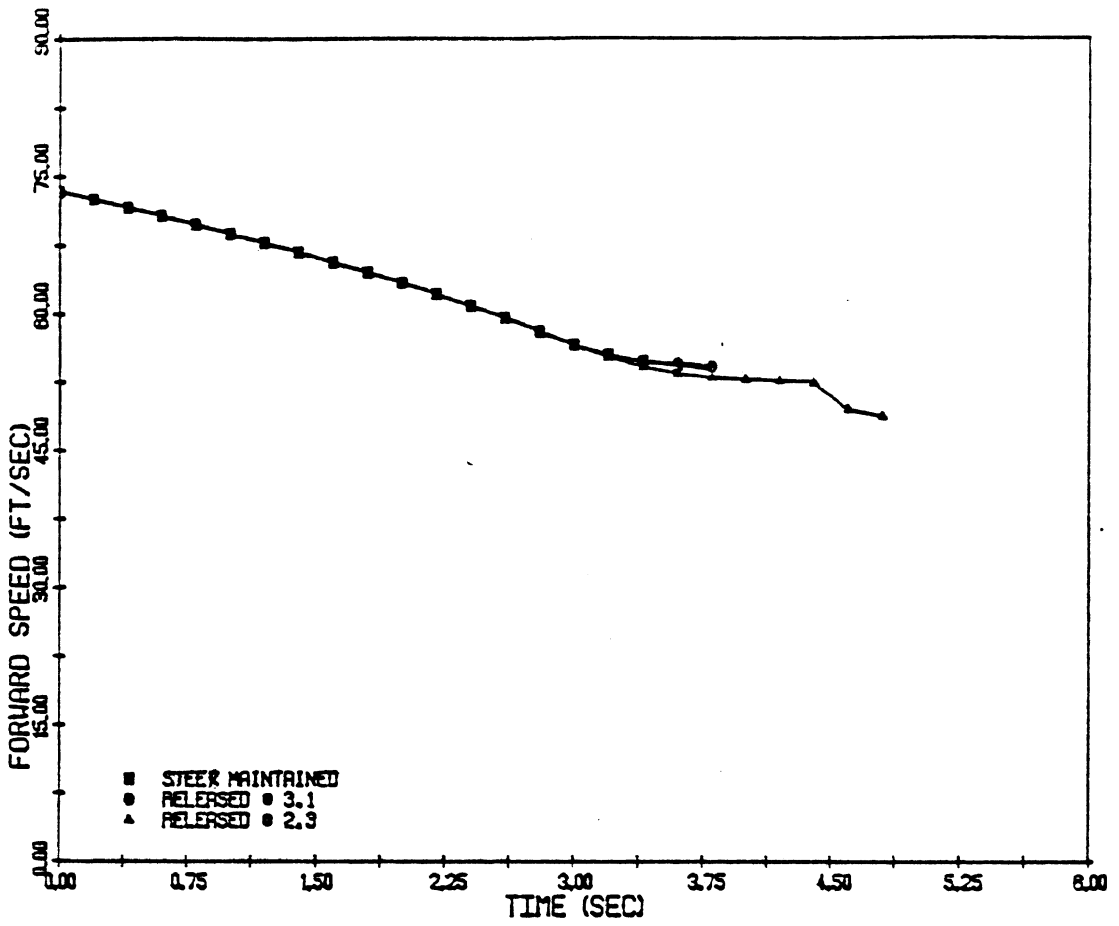
Figure 3(f)

30 and 70 mph. The results are shown in Figure 4. Steer angles of 10 and 13 degrees did not cause rollover from an initial 20 mph, and it appears likely that the vehicle cannot be rolled below this speed.

In the vicinity of the critical steer angle, the vehicle response levels reached are very sensitive to the input magnitude, especially at high speed. This behavior can be seen in Figure 3 by comparing the responses to 0.3 and 0.35 degrees of steer at 70 mph. This behavior is a property of vehicles which are understeer at low lateral accelerations and are oversteer at high lateral accelerations. Limiting oversteer is an undesirable response characteristic because of (1) its association with instability above a critical speed, and (2) the very high sensitivity of the responses to the input magnitude [29], [30].

In the case of the standard M-151 at 50 mph, a steer angle of 0.92 degrees is just sufficient to cause rollover in the J-turn. The roll angle reaches 56 degrees in 3.7 seconds, when all four wheels have lost contact with the ground. The effect of returning the steering input to zero at various stages of the maneuver have been studied briefly, and, in this marginal case, removing the input as late as 2.7 seconds into the run prevented rollover. The results are illustrated in Figure 5. If the original steer angle had been somewhat larger, so that this maneuver were not such a marginal case, it is likely that the subsequent removal of the input would be much less effective in preventing rollover.
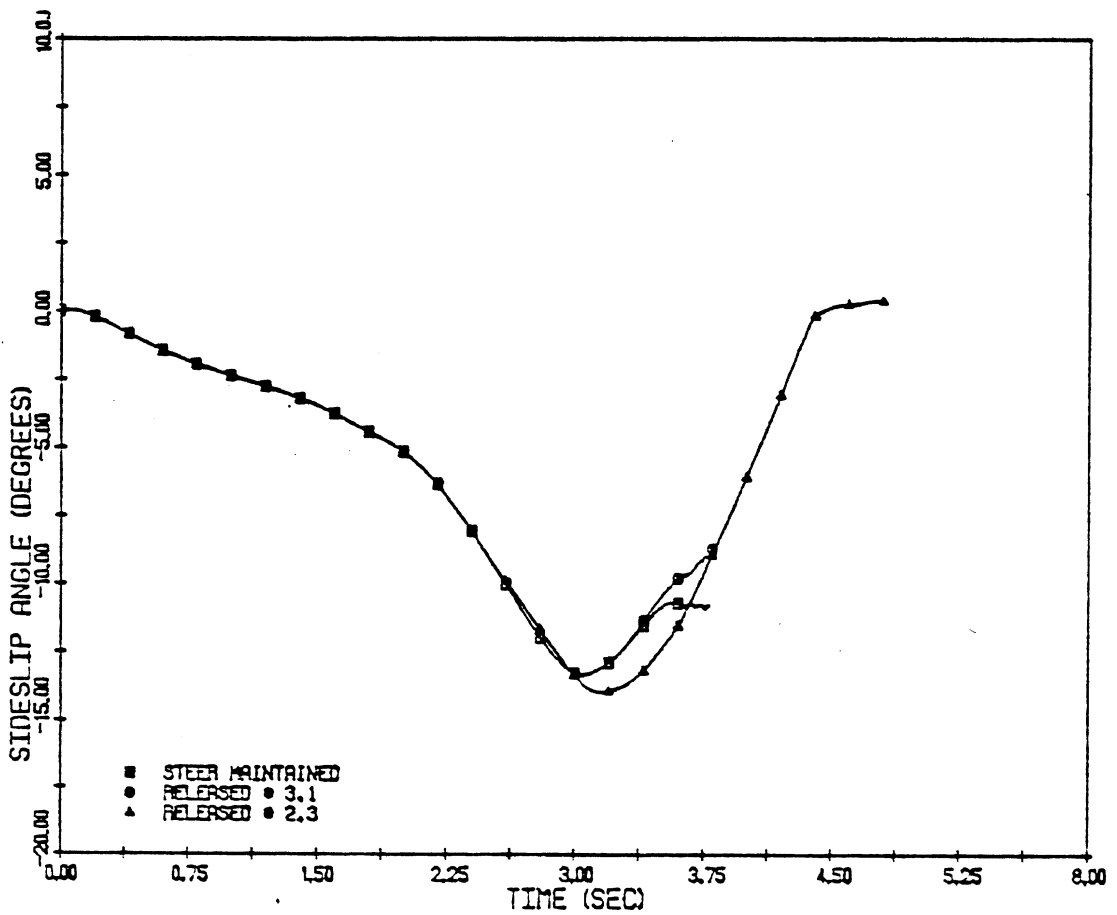
With increased loading of the M-151, typically involving the carrying of people in the rear seats and the recoilless rifle [27], the sprung mass increases, the mass center moves upwards and rearwards, the moments of inertia of the sprung mass are increased, and the equilibrium suspension positions are altered. Unfortunately, long-hand calculations are necessary when using HVOSM, to determine the modified input data representing the loaded vehicle. Such data were prepared (corresponding to the standard vehicle with the addition of two more passengers and the rifle), and the J-turn behavior of the loaded vehicle was examined at 50 mph. Steer input levels of 0.9, 0.93, and 1.1 degrees, which are near the critical level for the unladen vehicle, were employed. In each case, the vehicle rolled over. The event sequence was similar
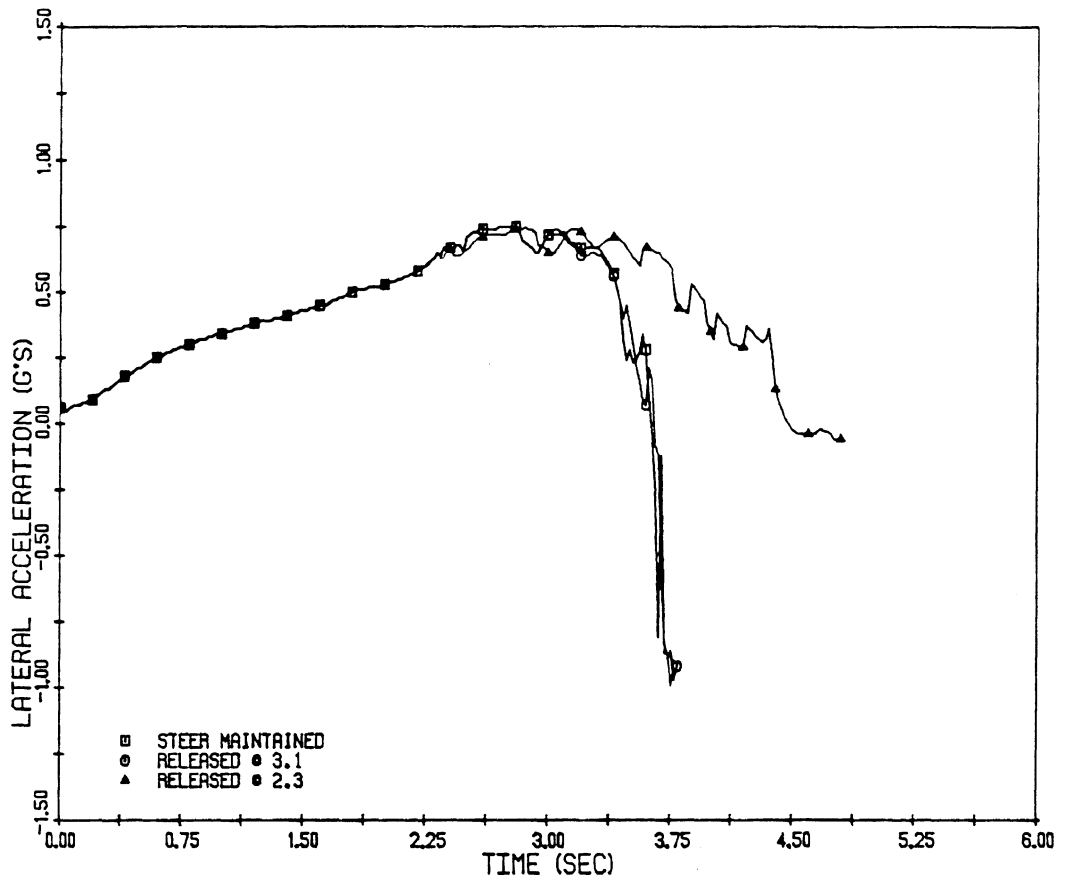
46 1512

Fig. 4 Computed Critical Steer Angle against Forward
Speed for M151 in T-turn Maneuver.

Forward speed, m.p.h.

Steer angle, degrees.

M151 SIMULATION, J-TURN, 0.92 DEGREE STEER, 50 MPH
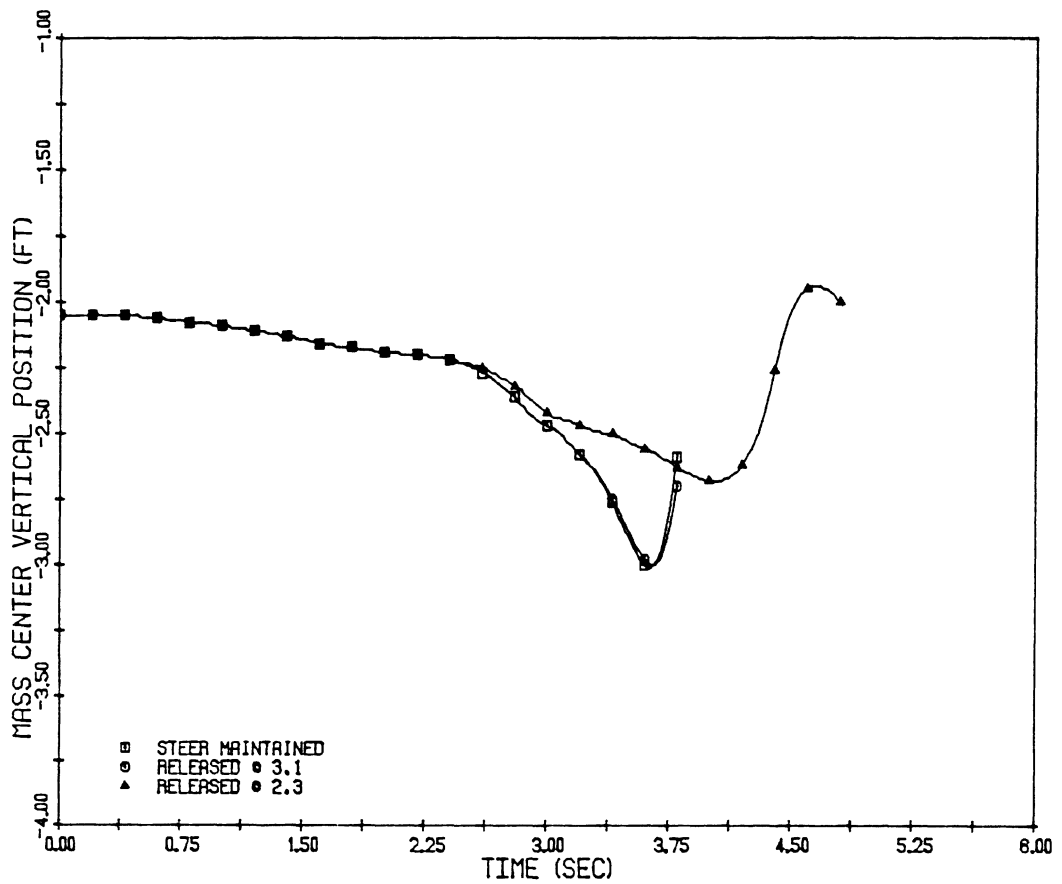
Figure 5(a)

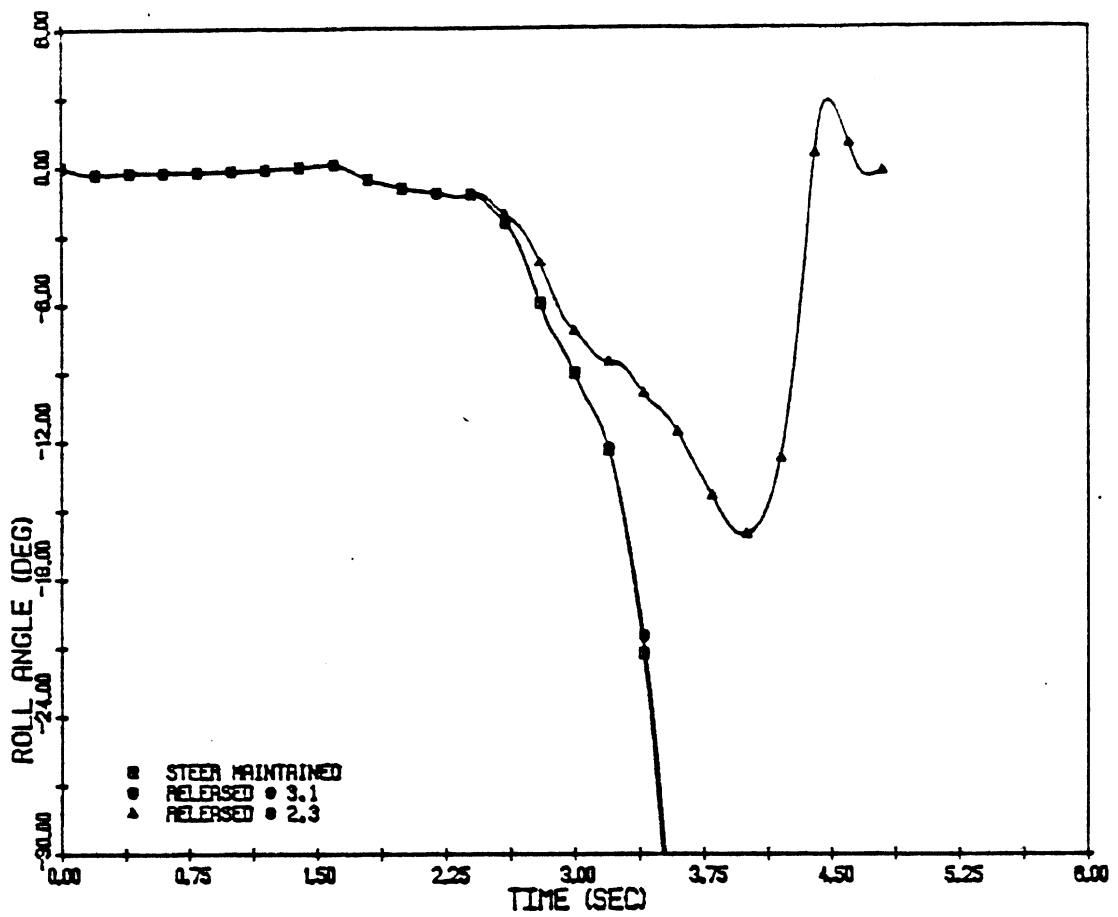M151 SIMULATION, J-TURN, 0.92 DEGREE STEER, 50 MPH

Figure 5(b)

42

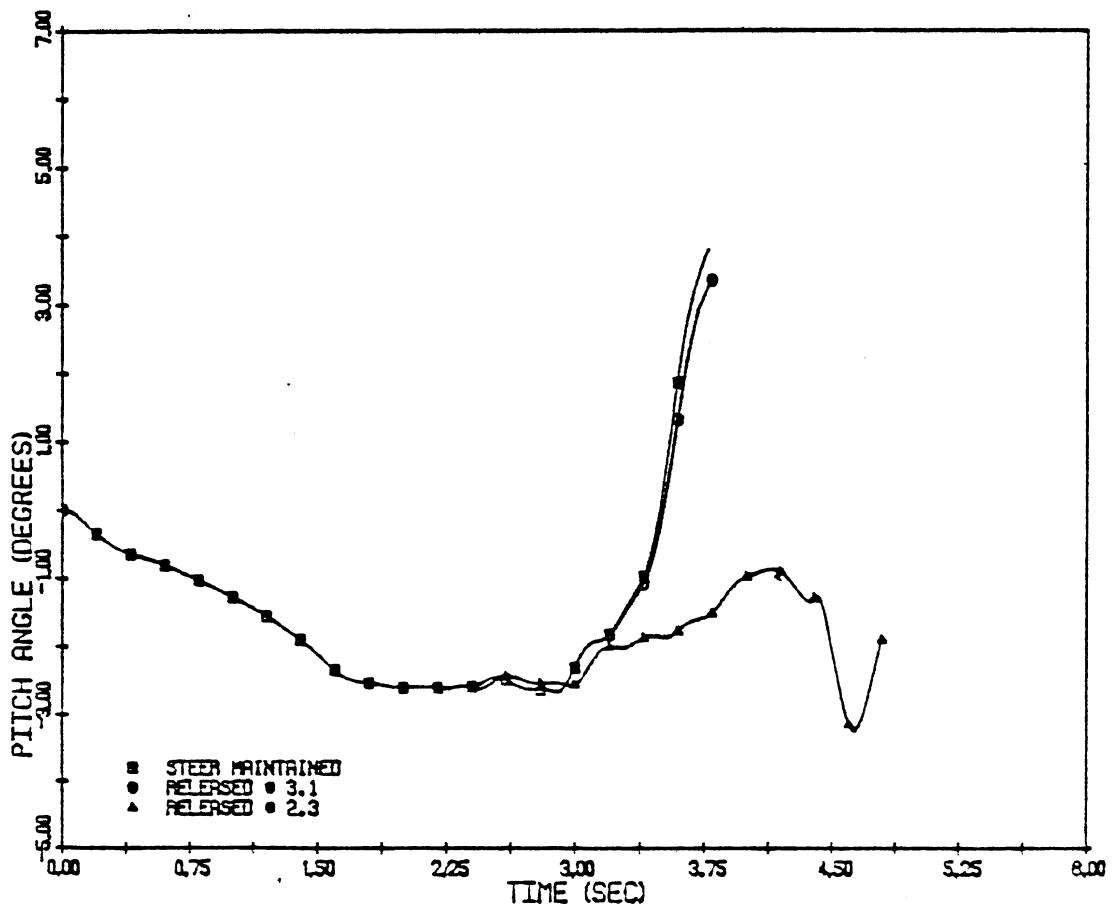M151 SIMULATION, J-TURN, 0.92 DEGREE STEER, 50 MPH

Figure 5(c)

M151 SIMULATION, J-TURN, 0.92 DEGREE STEER, 50 MPH

Figure 5(d)

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M151 SIMULATION, J-TURN, 0.92 DEGREE STEER, 50 MPH

Figure 5(e)

M151 SIMULATION, J-TURN, 0.92 DEGREE STEER, 50 MPH

Figure 5(f)

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to that described for the unladen vehicle, except that the pitch angle was noticeably slower in developing but achieved a greater magnitude (around 4.3 degrees) than in the "unloaded" case. The increased jounce displacements of the rear suspension deriving from the loading, causes the swing axle pivots to be lowered, which reduces the "jacking" tendency of the rear suspension. At the same time, more suspension movement before the rebound stops are contacted becomes possible. Thus the changes in the simulation results can be seen to be closely associated with the changes in the vehicle configuration. The behavior of the laden vehicle is shown in Figure 6.

In order to contrast the behavior of the M-151 with that of a more conventional vehicle, input data representing a 1963 Ford Galaxy were obtained using the preprocessing program and the HVOSM documentation (for the tire coefficients). J-turn maneuvers were simulated at 50 mph by applying steer angles of 2, 4, and 8 degrees to the Ford Galaxy. The results (see Fig. 7) show very little "jacking" or pitch of the sprung mass, comparative smoothness of the responses over time, an understeering type of response with sideslip angles of the front tires in general greater than those at the rear, leading to comparative insensitivity of the responses to the input at high input levels, greater load transfer across the front axle than the rear, and no suggestion of rollover even though lateral accelerations approaching 0.9 g were computed.

4.4   The Lane-Change Maneuver

A lane-change maneuver was simulated by assuming the standard M-151 to be traveling straight and level at 50 mph with one complete sine wave of steering input with a 2-second period being applied with the throttle closed at the commencement of the maneuver. Steer amplitudes of 0.5, 1.0, 2.0, and 4.0 degrees were employed.

The directional responses caused by the two smaller steer inputs were quite symmetrical with the vehicle being displaced laterally through roughly 4.5 and 9.5 feet, respectively. With the 2-degree steer input, the final path direction was yawed through 27 degrees in the direction of the initial steering input relative to the initial path, while with
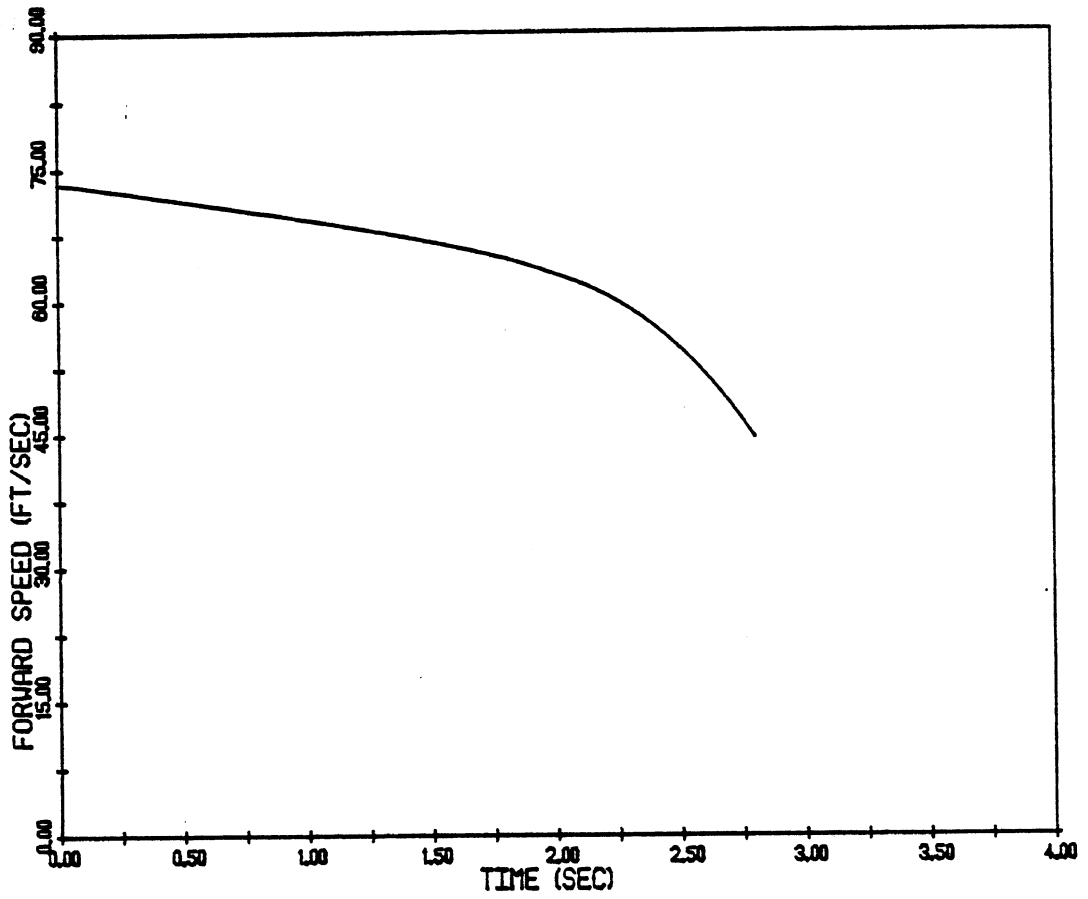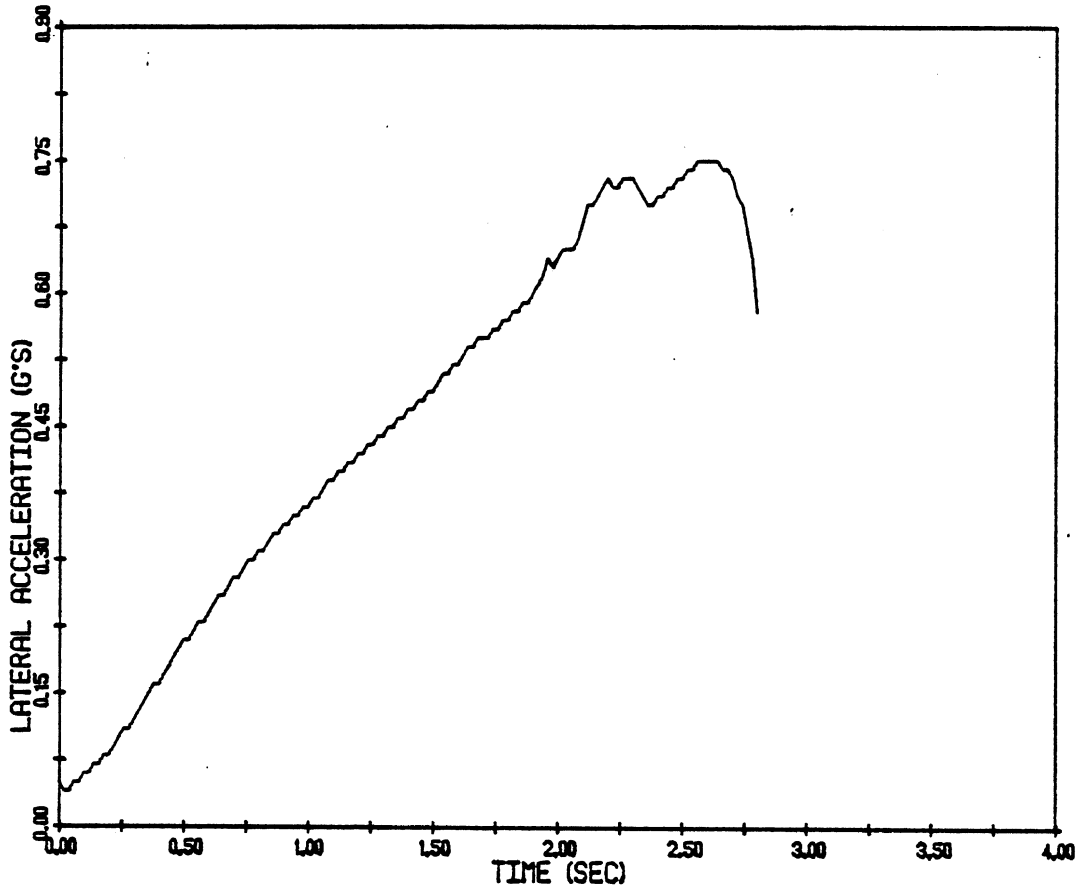
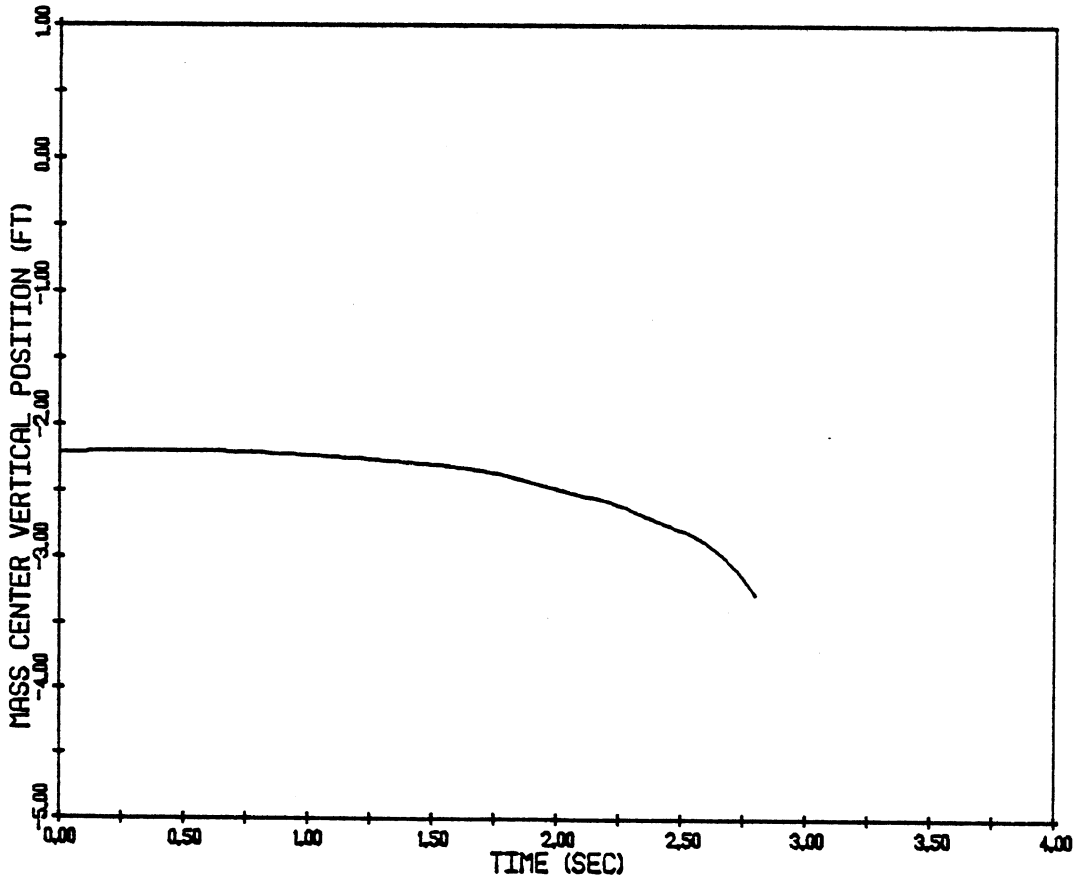M151 SIMULATION, LADEN, J-TURN, 0.9 DEGREE STEER, 50 MPH

Figure 6(a)
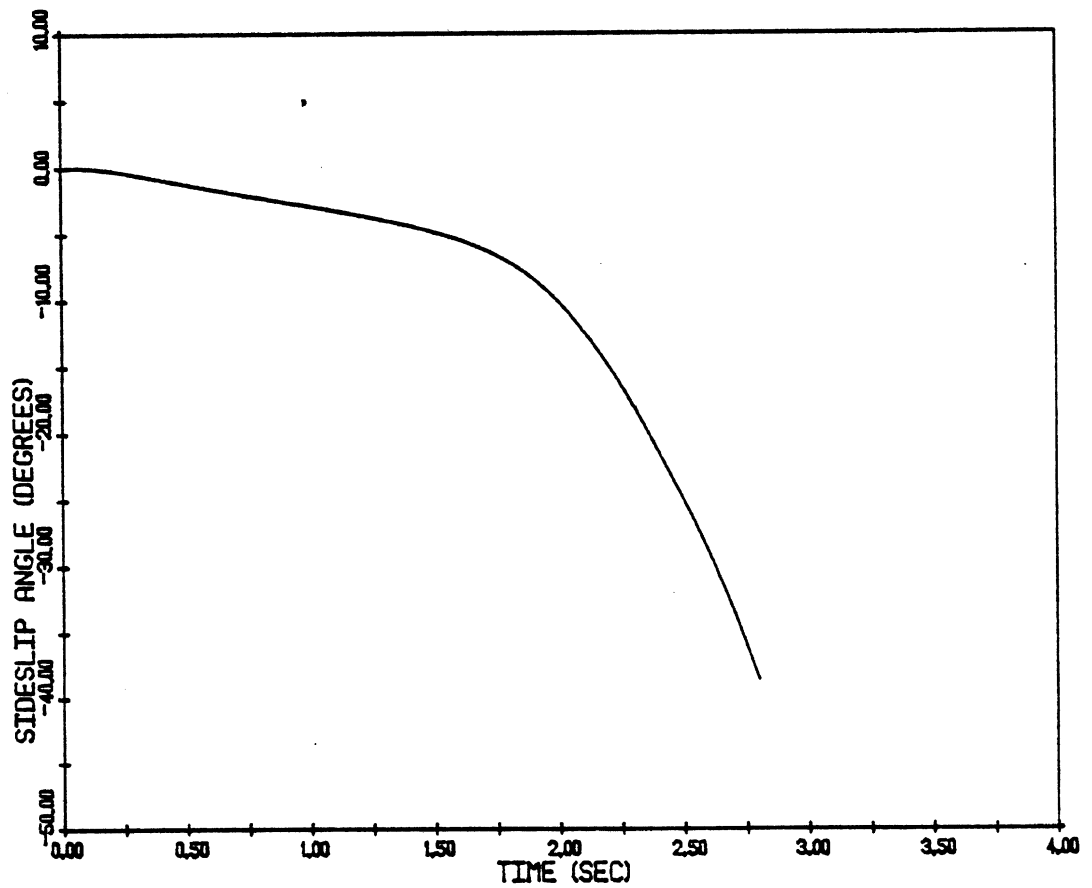
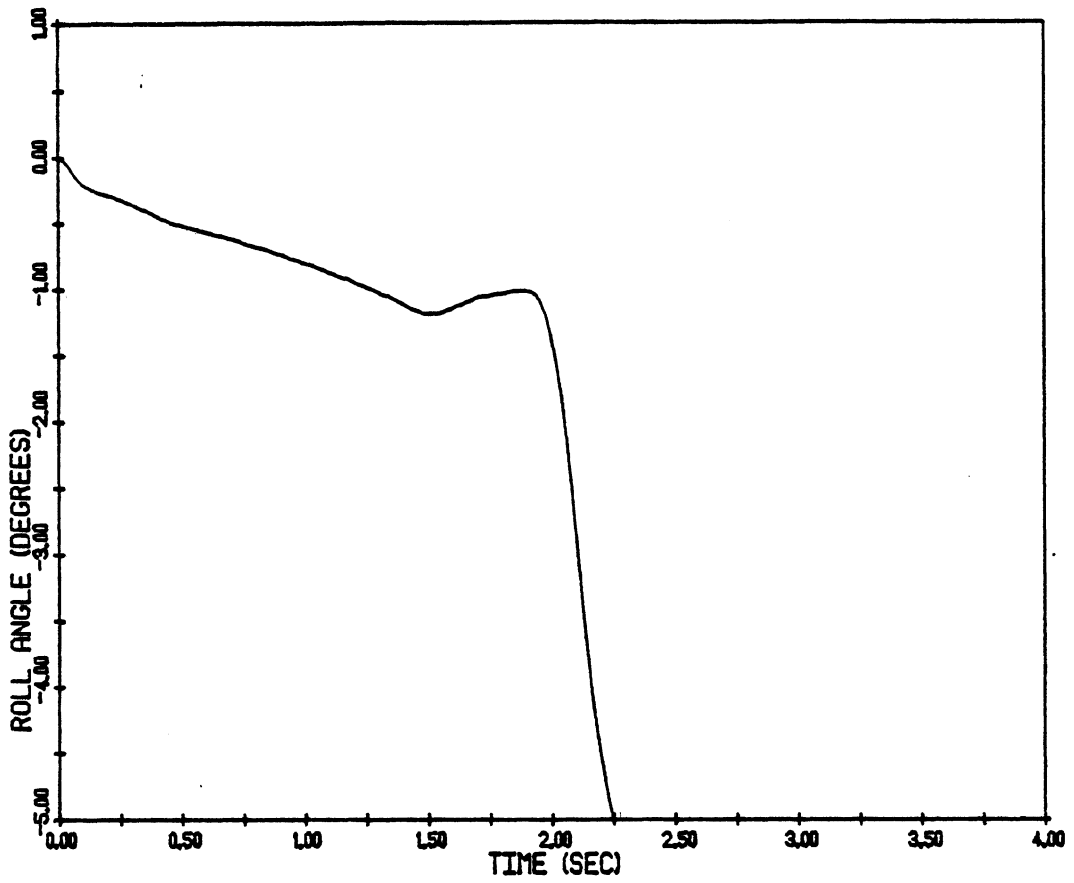M151 SIMULATION, LADEN, J-TURN, 0.9 DEGREE STEER, 50 MPH

Figure 6(b)

46

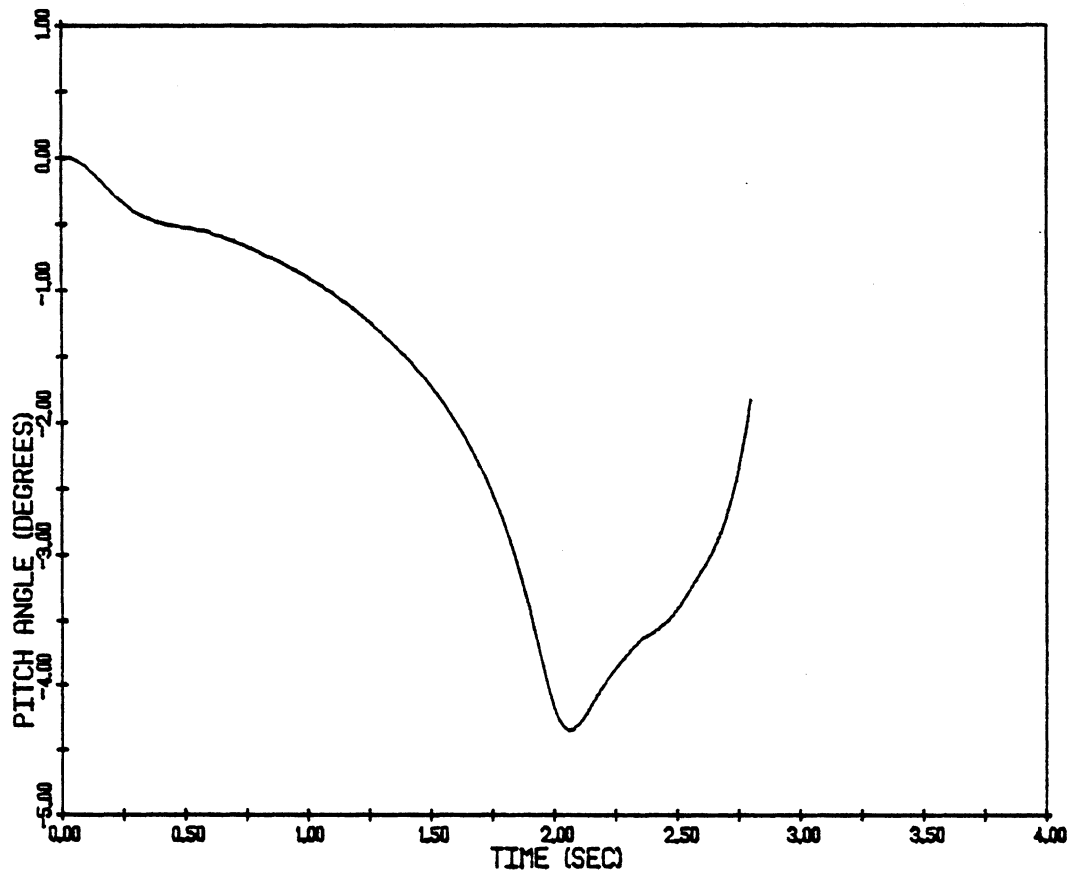M151 SIMULATION, LADEN, J-TURN, 0.9 DEGREE STEER, 50 MPH

Figure 6(c)

M151 SIMULATION, LADEN, J-TURN, 0.9 DEGREE STEER, 50 MPH
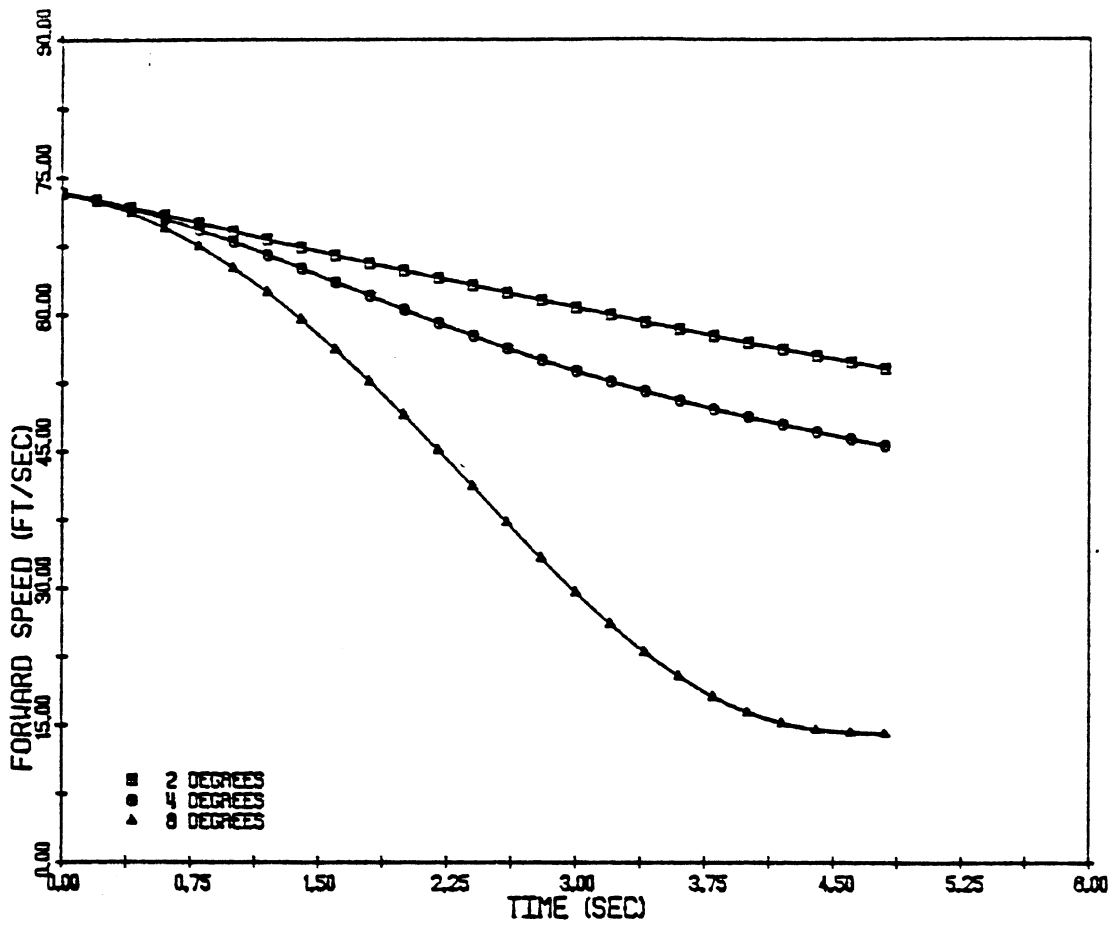
Figure 6(d)

47

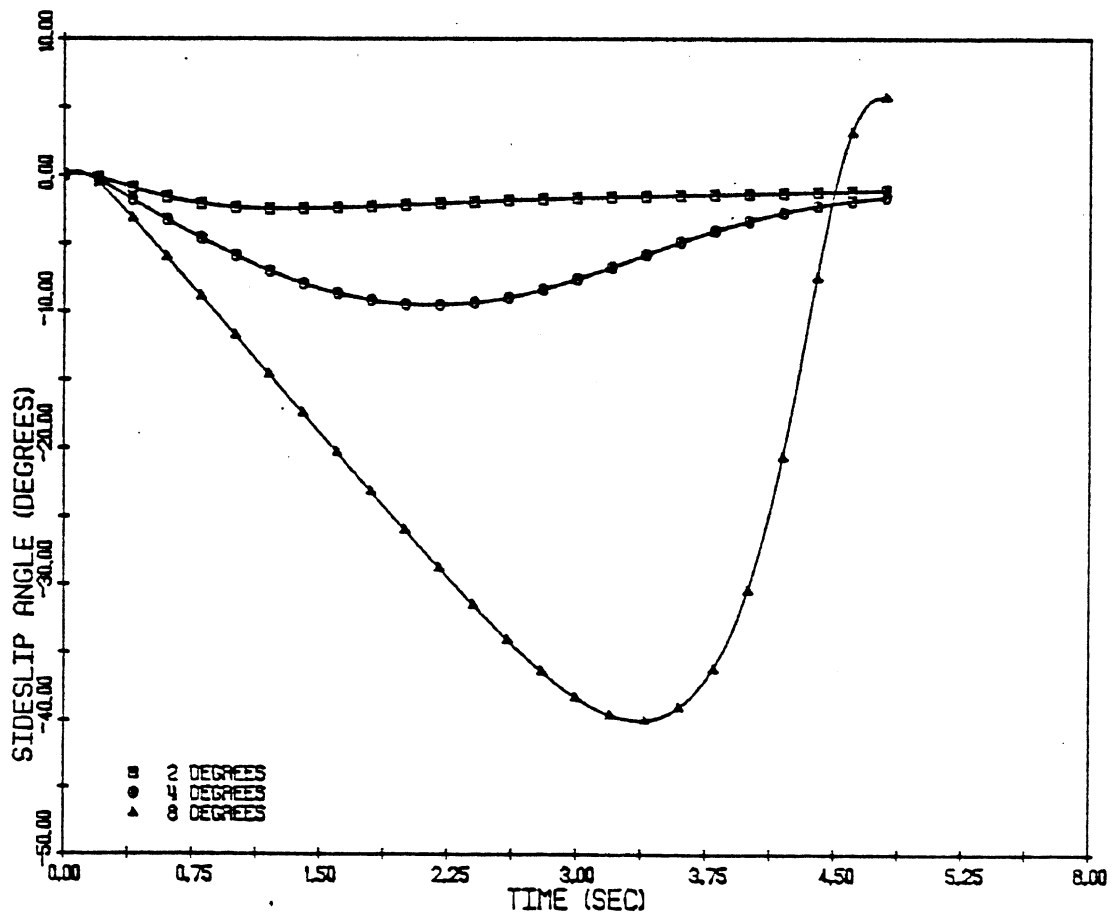M151 SIMULATION, LADEN, J-TURN, 0.9 DEGREE STEER, 50 MPH

Figure 6(e)

M151 SIMULATION, LADEN, J-TURN, 0.9 DEGREE STEER, 50 MPH

Figure 6(f)

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1963 FORD GALAXY SIMULATION, J-TURN, 2,4, AND 8 DEGREE STEER, 50 MPH

Figure 7(a)

1963 FORD GALAXY SIMULATION, J-TURN, 2,4,AND 8 DEGREE STEER, 50 MPH

Figure 7(b)
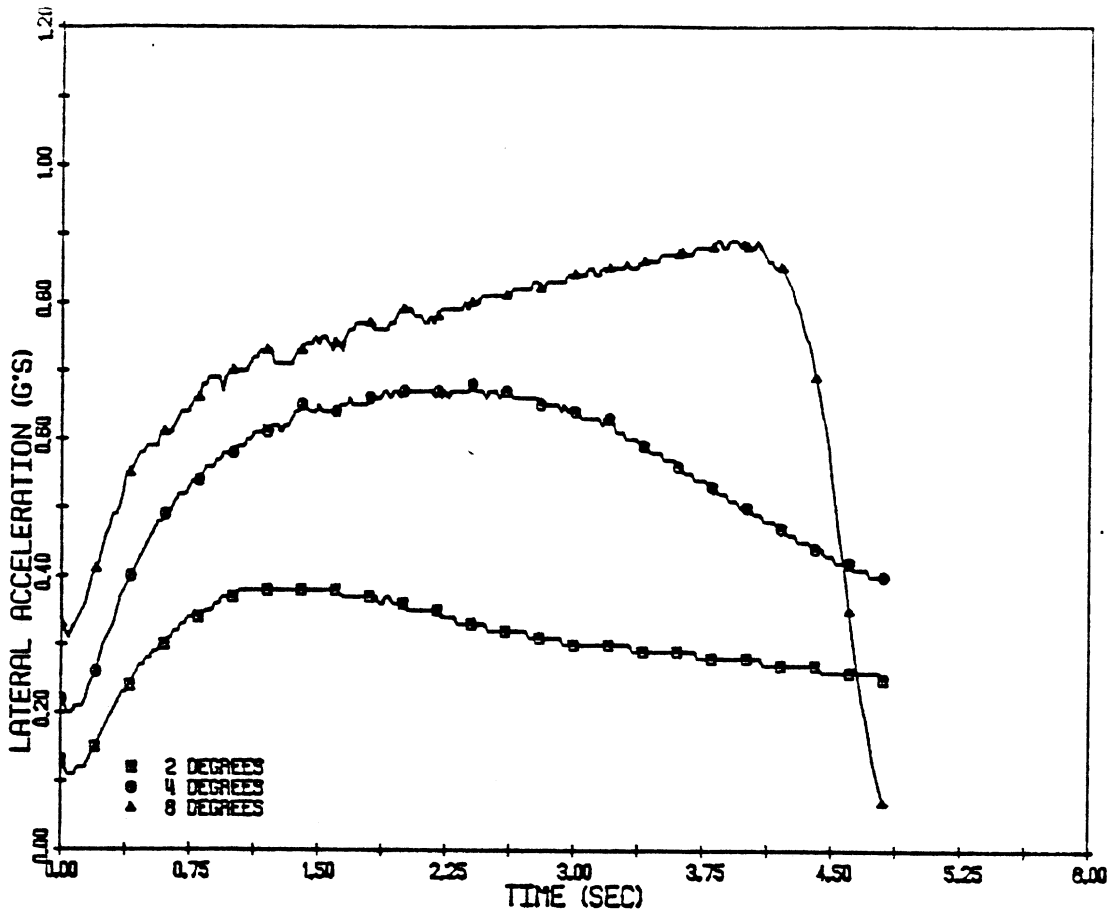
49

1963 FORD GALAXY SIMULATION, J-TURN, 2,4, AND 8 DEGREE STEER, 50 MPH

Figure 7(c)

1963 FORD GALAXY SIMULATION, J-TURN, 2,4, AND 8 DEGREE STEER, 50 MPH

Figure 7(d)

50
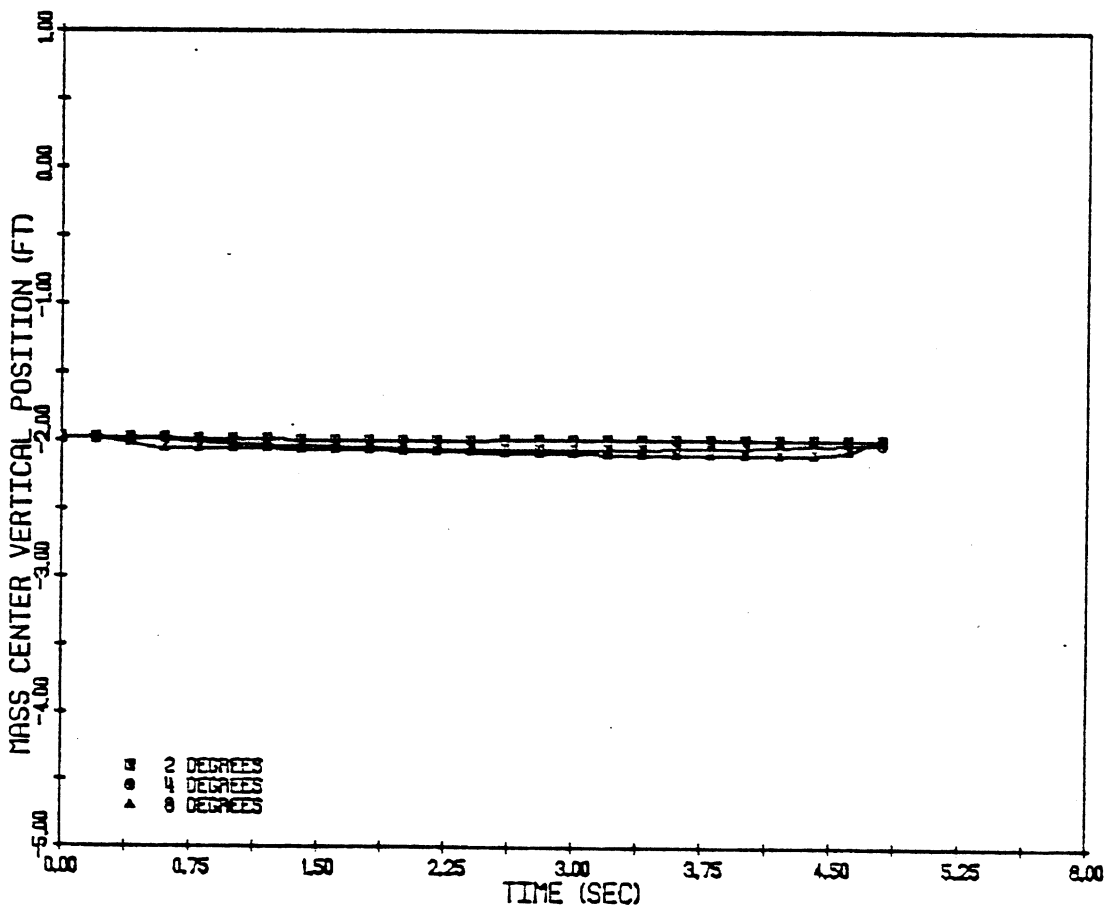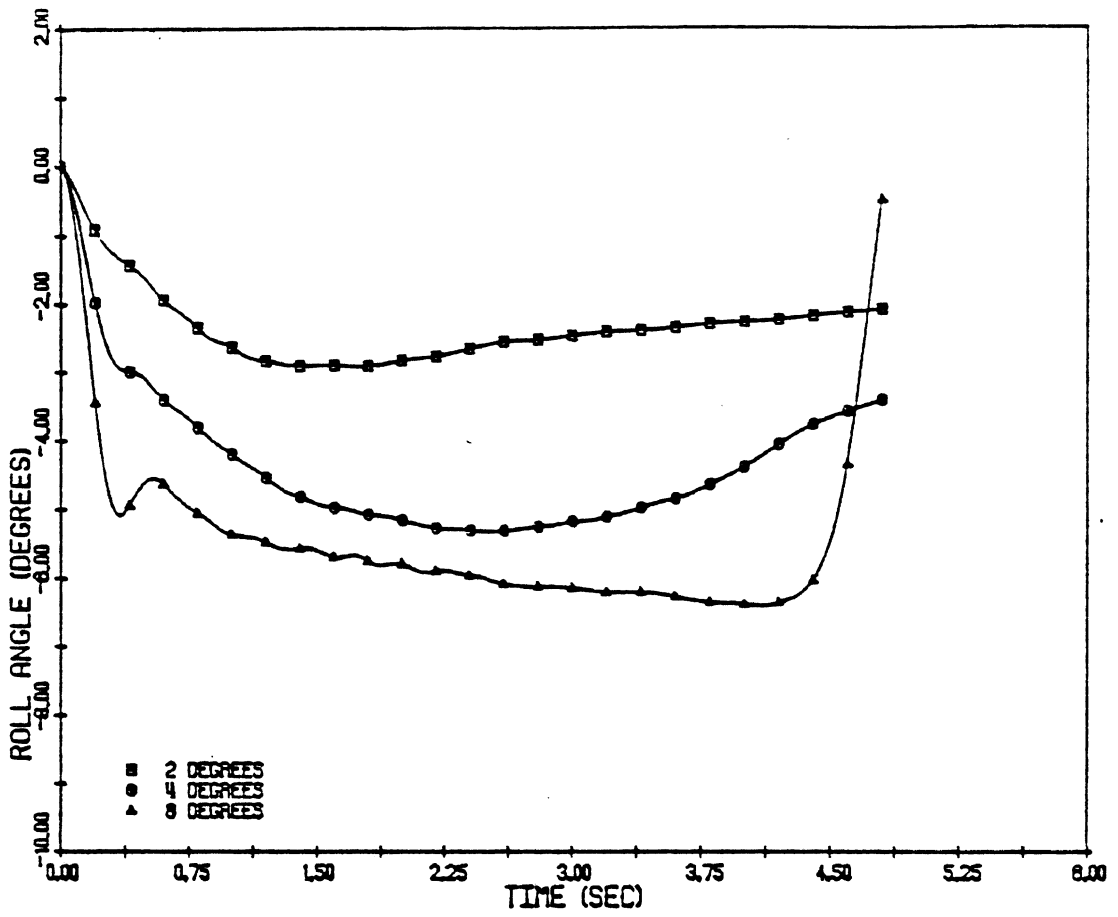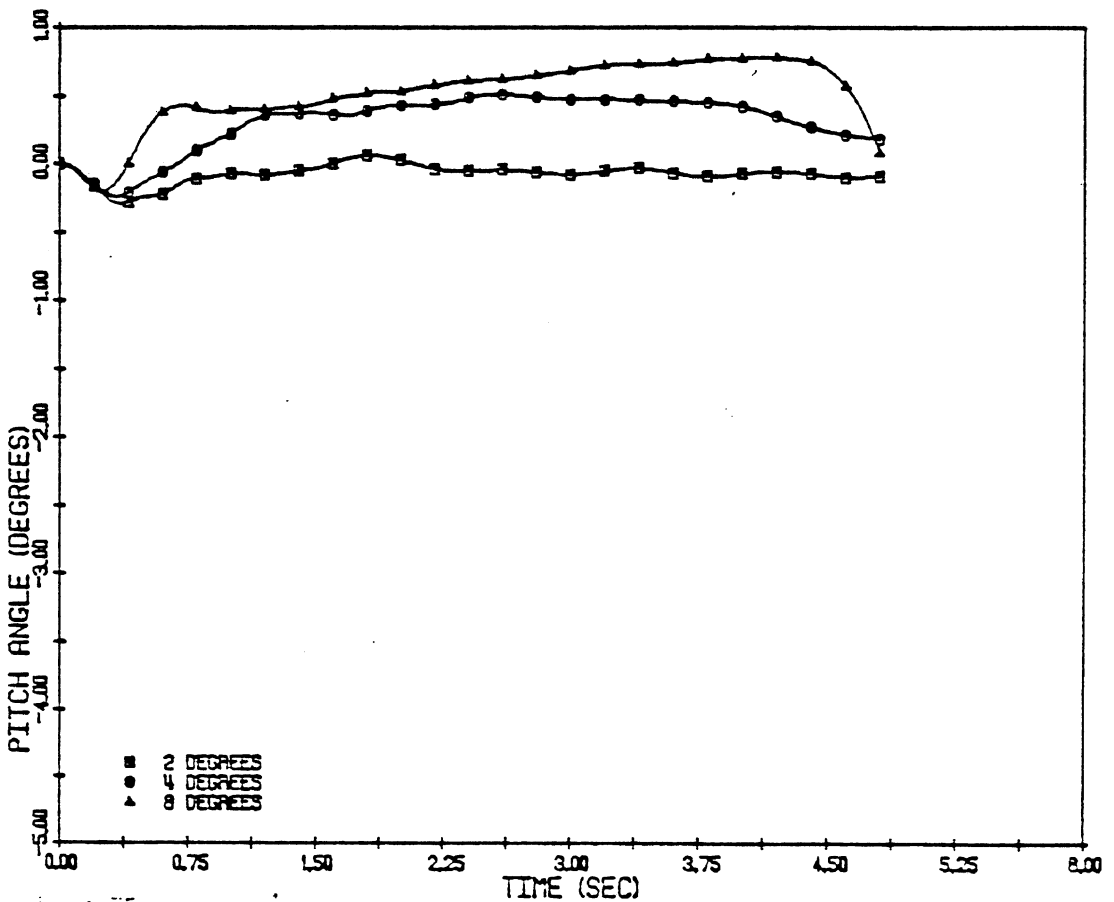
1963 FORD GALAXY SIMULATION, J-TURN, 2,4, AND 8 DEGREE STEER, 50 MPH

Figure 7(e)

1963 FORD GALAXY SIMULATION, J-TURN, 2,4, AND 8 DEGREE STEER, 50 MPH

Figure 7(f)

the 4-degree input, the vehicle rolled over in response to the first quarter of the sine wave.

The results show that the two smaller inputs produced very mild maneuvers, with the lateral motion responses for the 1-degree input being almost double those for the 0.5-degree input. This was not true for the pitch angle response, part of which is caused by closing the throttle, with the remaining portion of the response being caused by very nonlinear effects such as lateral load transfer, the nonlinearity of the tire side force versus load, and the changing suspension kine- matics when "jacking" takes place. A maximum lateral acceleration of 0.51 g was reached for the 2-degree input, and the inside rear wheel almost lifted at one point. The maximum roll angle was just over 1 degree. These results are shown in Figure 8.

Further study of the lane-change maneuver involving the examina- tion of input levels between 2 and 4 degrees and periods of sine-wave steering other than 2 seconds would obviously be possible. However, such study was not considered to be very cost effective in providing an understanding of M-151 behavior over and above that obtained from the J-turn and combined steering and braking simulations.
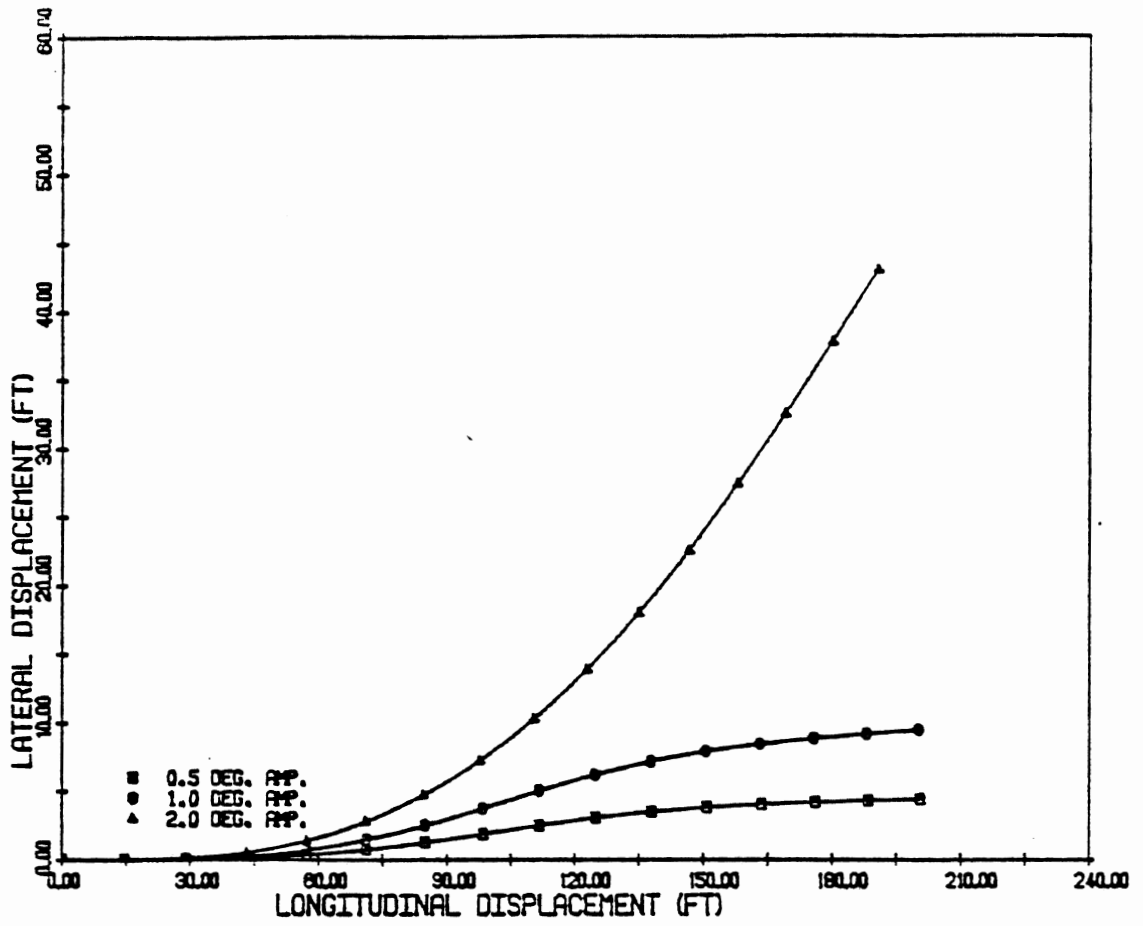
M151 SIMULATION, LANE CHANGES, 50 MPH

Figure 8(a)
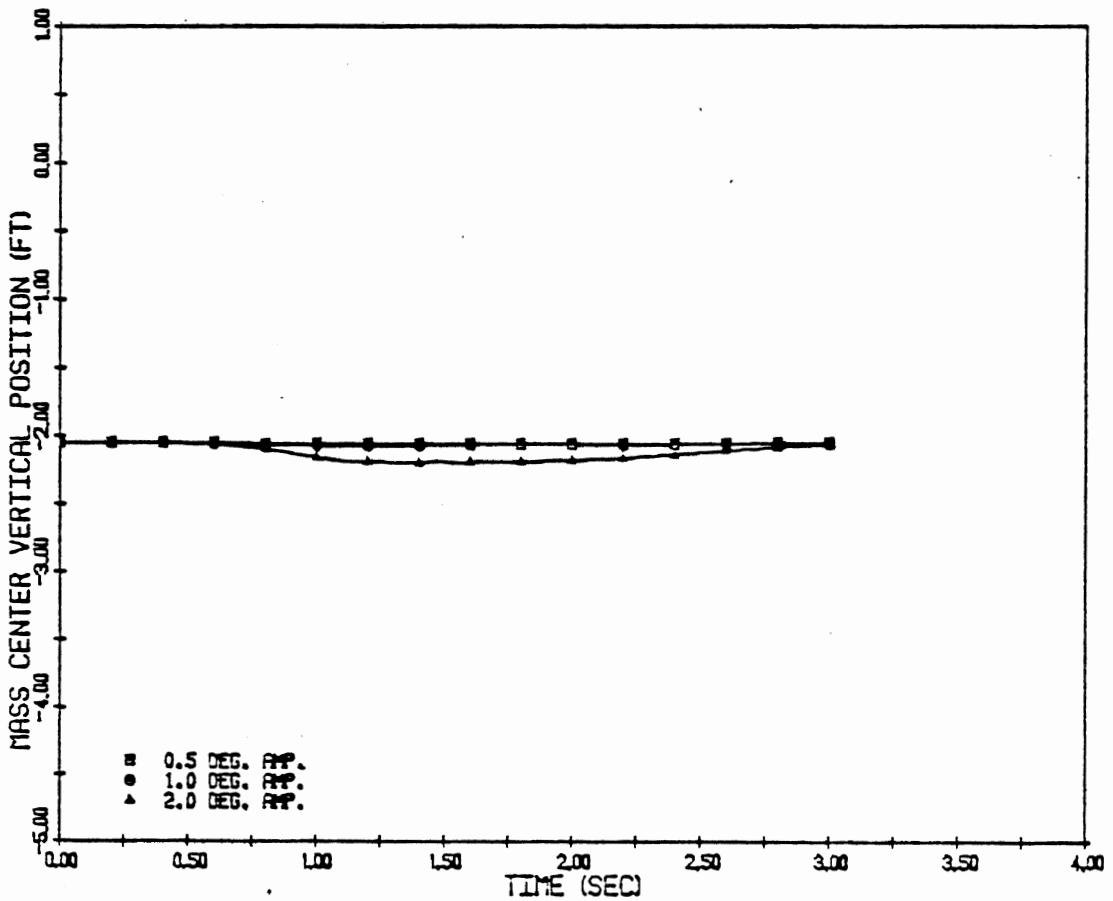
M151 SIMULATION, LANE CHANGES, 50 MPH

Figure 8(b)

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M151 SIMULATION, LANE CHANGES, 50 MPH

Figure 8(c)

M151 SIMULATION, LANE CHANGES, 50 MPH
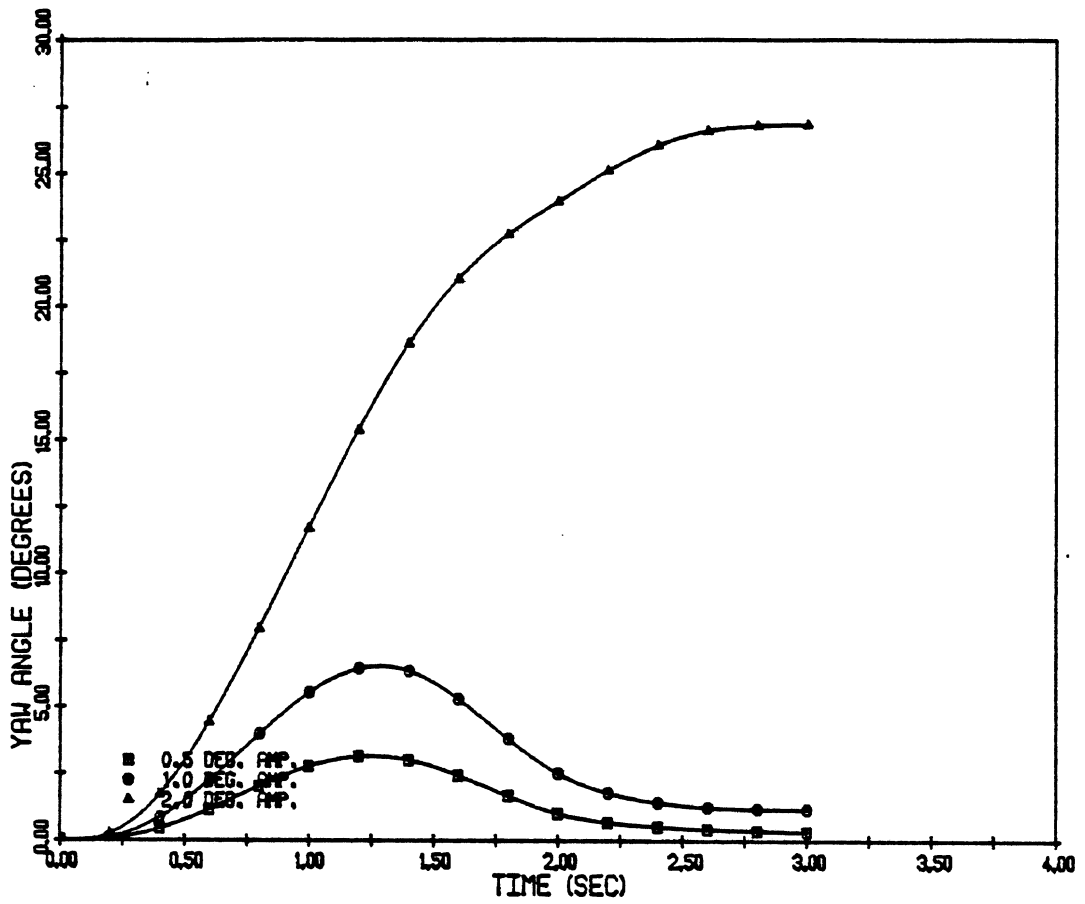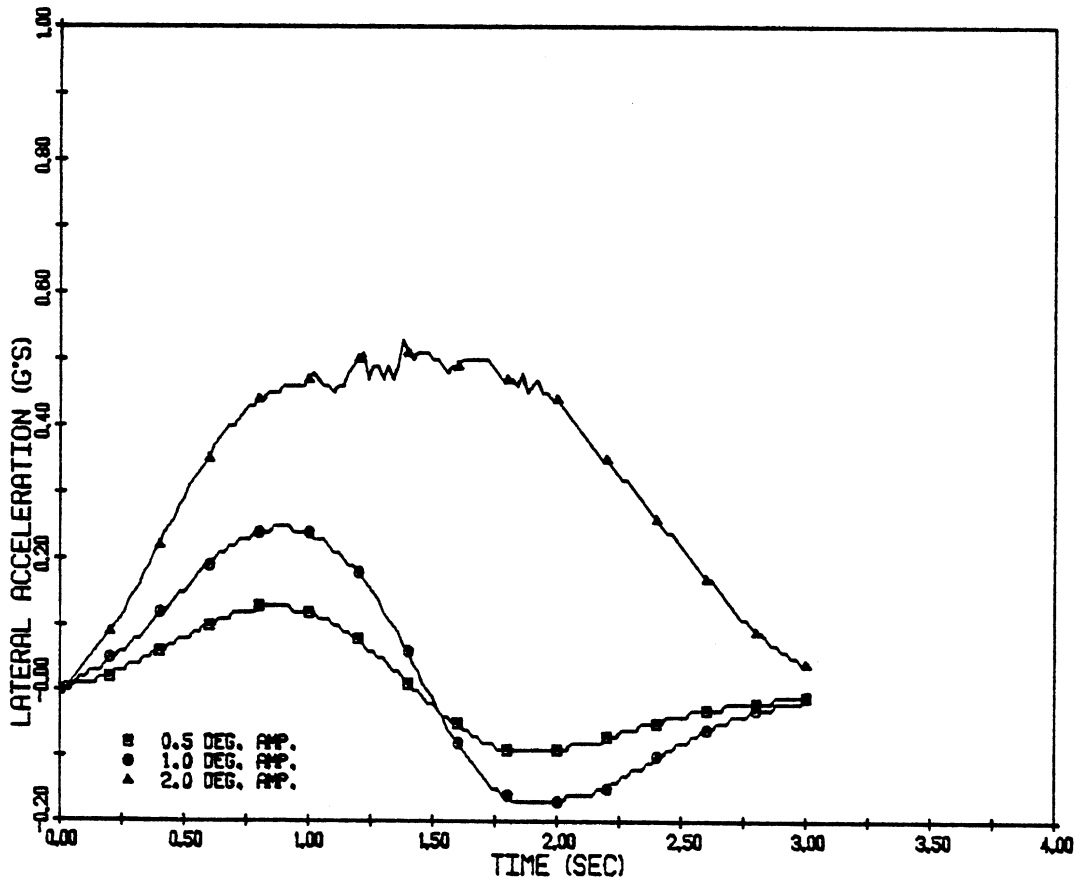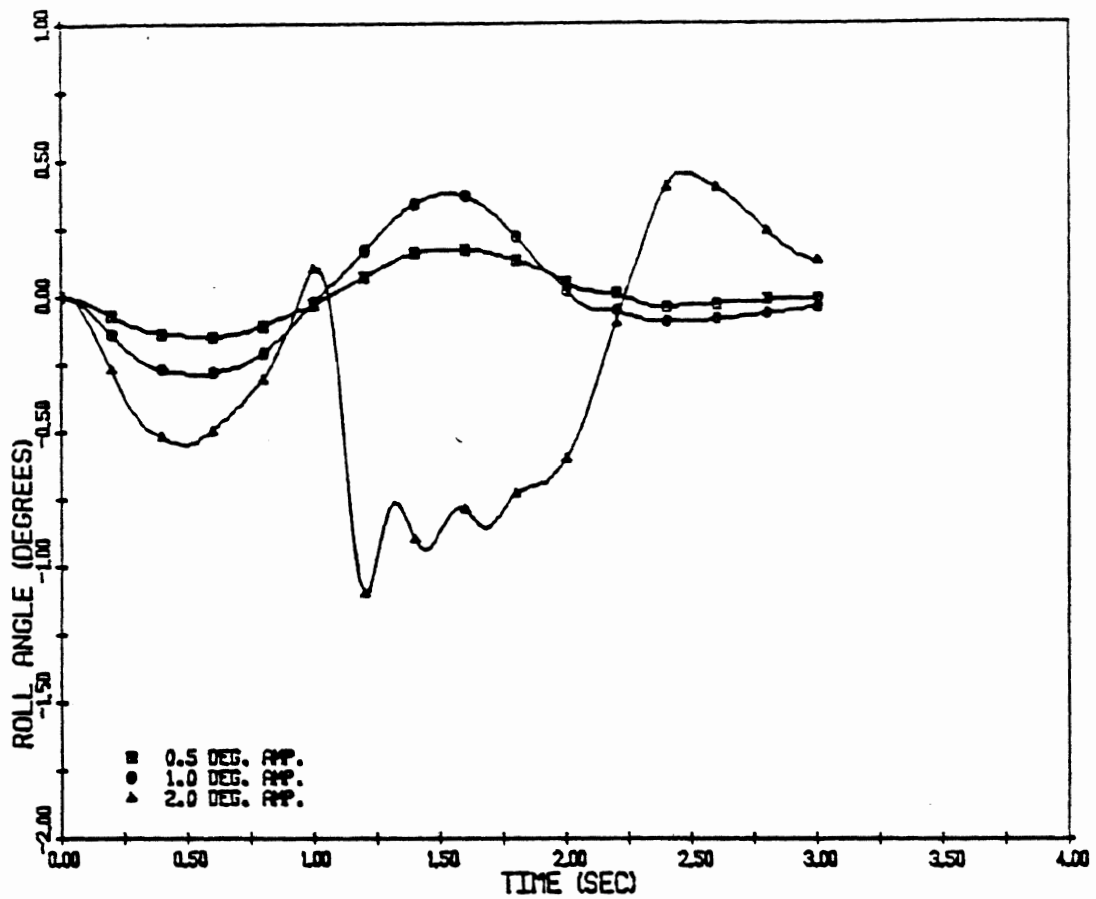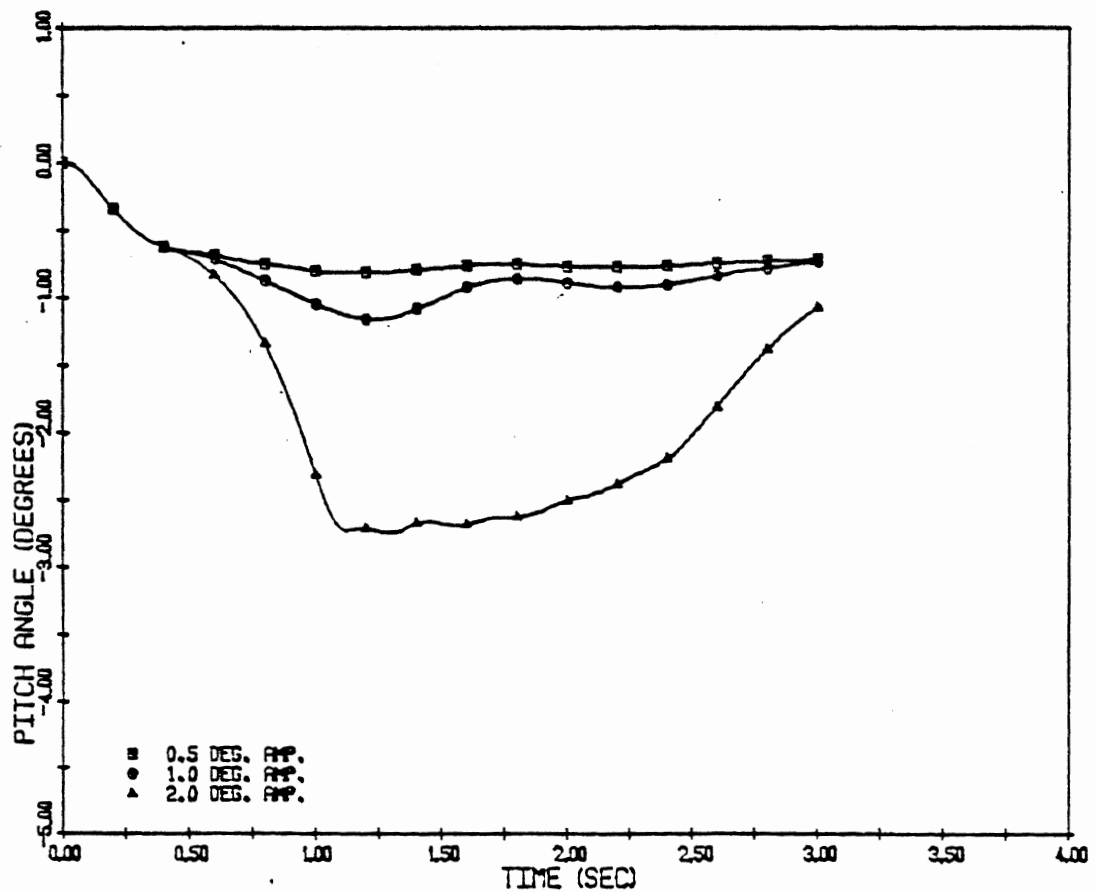
Figure 8(d)

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M151 SIMULATION, LANE CHANGES, 50 MPH

Figure 8(e)

M151 SIMULATION, LANE CHANGES, 50 MPH

Figure 8(f)

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5.0 FINDINGS

5.1 The Rollover Threshold in J-Turn Maneuvers

As described earlier, for each vehicle speed there is a critical steer input above which the vehicle will rollover and below which it will not. This finding applies, of course, to the standard M-151 on a high friction surface. The relationship between speed and critical steer angle is shown in Figure 3.

In order to establish the minimum speed at which rollover could be provoked in a J-turn, a steer angle of 30 degrees was employed as the control input in some low speed runs. This action produced an unexpected result, namely, that a 30-degree steer input will not cause rollover at speeds between 30 and 40 mph, although much smaller steer inputs will. This finding is considered, however, to be an artifact since, in real life, a 30-degree input is impractical and could not be applied in a stepwise fashion. With this large steer input, the sequence of vehicle motions is substantially different from that resulting from inputs near the critical value, and close examination of the results shows them to be physically reasonable. Also, in one 40-mph run starting from a quasi-equilibrium attitude with the throttle open, the M-151 rolled over with a constant 1-degree steer angle, which is half the critical value for the J-turn at that initial velocity. Again, the character of the motions was different from that typical of a J-turn, and the importance of force and motion phasing effects in determining the input levels at which rollover will occur is indicated.

The J-turn results have been examined in detail to determine whether or not there are marked differences between the motions leading to rollover and those near the limit which do not. Lateral acceleration levels, vehicle sideslip angles, and suspension behavior are very similar until rollover is imminent. Characteristically, the roll angle increases very rapidly when rollover occurs, and the corresponding roll velocities tend to be significantly higher than those occurring in non-rollover runs. However, these high roll velocities begin within a half a second of rollover, by which time the roll angle itself would be

sufficient to indicate to a driver that rollover was imminent. Lifting
of the inside rear wheel from the pavement could provide early warning
of near-limit conditions, typically preceding rollover by about 1 second
and coinciding with a lateral acceleration of the sprung mass center
of gravity in the region of 0.65 g. Such a condition, however, although
necessary, is not sufficient to produce rollover in that the inside
rear wheel also lifted in many non-rollover simulations.

5.2   The Influence of Loading

As discussed earlier, J-turns have been simulated with the M-151
in one characteristic, heavily laden condition (viz., carrying two
rear seat passengers and the recoilless rifle). Rollover occurs at lower
steer input levels than for the standard loading condition, with sub-
stantially the same event sequence except for expected variations in
the pitch angle response. Although heavily laden, the rear roll center
is still high enough for "jacking" to be very powerful at high lateral
accelerations. Once the "jacking" is under way, the rear suspension
geometry tends to revert to that of the unladen vehicle (i.e., the
roll center is raised). The inside rear wheel lifts off the pavement
about 1 second ahead of rollover, when the lateral acceleration of the
mass center is again near 0.65 g.

5.3   Comparison of M-151 and Ford Galaxy Behaviors

The simulation results are consistent with our expectation that
the Galaxy would understeer throughout the lateral acceleration range.
With steering input levels which keep the vehicle's lateral acceleration
below about 0.3 g, its responses are linear functions of the input,
but for larger steer inputs, the output/input ratios decrease. For a
given vehicle speed, these gain changes are a very regular function of
steer angle. A little front-suspension jacking occurs, but is not
sufficient to have much influence on the vehicle's lateral motions. This
understeer behavior derives from (1) the mass center being nearer the
front axle than the rear axle, (2) the major part of the lateral load
transfer taking place across the front tires, (3) the front wheels in-
clining substantially with the body while the rear wheels remain upright,

and (4) the front tire pressures being lower than those at the rear. The small disturbance stability of the Ford Galaxy, together with the uniformity of its transition in response behavior from the low to the high lateral acceleration regimes, make it relatively easy to control by a driver. Its relatively low c.g.-height-to-track-width ratio also make it stable in roll.

The M-151 behaves in a manner similar to the Galaxy at low lateral acceleations, but changes towards oversteer as the lateral acceleration increases. Up to a point, the roll attitude changes very little with cornering, but then, as the load transfer across the rear wheels encourages "jacking," and as the "jacking" feeds on itself, lifting the vehicle mass center to further increase load transfer and also lifting the rear roll center, the vehicle response behavior changes character with relative rapidity. Particularly for high speeds, the vehicle responses at higher levels of lateral acceleration become very sensitive to the input, and the instability of the open-loop vehicle for small disturbances from usual operating conditions may well be common. In these circumstances, the driver is required to provide stabilizing feedback control. Controlling the M-151 near its cornering limit, particularly at high road speeds, would seem to be a relatively difficult task, and this fact, together with its high mass center and narrow track, and the rear suspension "jacking" which it suffers, make it likely that it will be involved comparatively frequently in rollover type accidents.

The fact that the inside rear wheel of the M-151 loses contact with the road at a lateral acceleration close to 0.65 g has an important bearing on its response behavior. It will not maintain a steady condition above this lateral acceleration, because, when the wheel lifts, drive thrust is not transmitted to the road, and the vehicle slows down. If the vehicle speed is low (below 25 mph, say) at the start of a maneuver, the indications from the simulation results are that a practically achievable rate of application of steering input will not lead to rollover because of the wheel lifting and loss of forward speed. The very low speed test course at the Aberdeen Proving Ground [24], with its fairly gentle turn entry and exit profiles, would not be

expected to provide a good test of the M-151's rollover potential.
The simulation results, in fact, agree qualitatively with the obser-
vations recorded in Reference [24]. If high speed testing were to be
attempted, a driver could not be expected to fully explore a vehicle's
roll stability without considerable measures being taken to ensure his
safety. It appears that, at minimum, these measures should include
outriggers becoming effective at (say) 20 degrees of roll and a large,
obstruction-free test area.

6.0 PROSPECTS FOR THE SYNTHESIS OF A ROLLOVER INDEX

Before addressing the prospects for synthesizing a "rollover index," it is essential that we define what is meant by "rollover index." Such a definition is not a straightforward matter and, to a very large extent, will depend on the interpretation of the vehicle rollover problem being encountered by the U.S. Army. Accordingly, this section of the report will be prefaced with some discussion of the "problem" that the U.S. Army would like to alleviate, to the extent that there are practical and cost-effective means for doing so.

The "problem" can, of course, be defined in a number of different ways. Assuming that there is clear, unambiguous evidence showing that rollovers occur in over-the-road operations of some military vehicles to a much greater extent than others (when the data are properly normalized for exposure), questions can be raised as to "Why is this the case?" and "What are the corrective measures that should be instituted?"

With respect to the first question, the answer could be either in the design or operational realm. For example, it is clear that the design choice made with respect to the suspension geometry employed on the M-151 led to a highly effective off-road vehicle having a host of desirable off-road qualities at the expense of obtaining a vehicle which becomes somewhat hazardous when (1) driven at speed over paved road surfaces and (2) the driver encounters some emergency whose resolution leads to "emergency" steering and braking control actions. Whether this design tradeoff was fully understood at the time that the M-151 was first being developed by the Army is not known. Nevertheless, at this point in time, it is important that the Army fully appreciate the tradeoff that is involved since a complete systems analysis might indicate that the "rollover problem" should be addressed in the operational realm rather than in the design realm.

In contrast to the M-151 which uses independent suspension-system designs that compromise its emergency maneuvering characteristics on

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hard, dry pavement, the Army also uses a large number of solid-axle trucks in both the goer and non-goer configuration. Such vehicles generally exhibit a rollover threshold prior to reaching their cornering limit and this behavior can, in theory, be looked upon as a price that must be paid if it is necessary to transport military material in a reasonably productive manner.* In this instance, it is necessary that those who are responsible for vehicle procurement and those who are responsible for its operation clearly understand that the manner in which the military truck is loaded and used can (and does) influence its rollover threshold. With respect to the second question, the experience obtained in the commercial-trucking enterprise indicates that the alternatives for raising the rollover threshold of trucks are limited in number. Whereas attention to design detail is certainly warranted in the case of both commercial and military trucks, it appears that it is primarily the user who must be on the lookout for loading and usage practices that reduce the rollover threshold below some nominal and, presumably, unacceptable level.

The simulation findings presented above suggest that, in the case of a vehicle that becomes both directionally and roll unstable at higher levels of lateral acceleration, it is not straightforward to establish definitive conditions under which the vehicle will either rollover or not rollover. In general, it was established that this particular vehicle cannot be brought to a limiting steady value of lateral acceleration beyond which all four wheels will leave the road as the vehicle rolls. Rather, a dynamic maneuver is required to cause rollover on a smooth, level surface. Further, it became clear that to the extent that a "step-steer" maneuver is a realistic maneuver approximating driver action in an emergency, then there is a boundary in the "speed-steer displacement space" which says that values of speed and steer input exceeding a certain limit value will result in rollover, whereas values of speed and steer input below that limit will not cause a rollover event. It should be noted that the boundary defined by Figure 4 does not exist for many four-wheeled passenger vehicles in

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*This observation is equally valid for the commercial motor truck and truck combination.

that they cannot be caused to rollover on a realistic high friction surface. On the other hand, vehicles such as the M-151 possess such a boundary with the location of this boundary being sensitive to its loading condition. Further, goods-carrying vehicles, such as single-unit trucks and tractor-trailer combinations, will have a boundary that is maneuver sensitive as well, since they can be rolled over by marginally exceeding a quasi-static threshold as well as being rolled over as a result of some transient maneuver.

To the extent that one is willing to associate the concept of a rollover index with that of a rollover threshold, it becomes possible to state that the curve presented in Figure 4 constitutes a rollover index for the M-151 in one particular loading condition. It should be clear that a different curve defines the rollover index of the M-151 in a different loading condition. Further studies, both analytical and experimental, would be required to determine the extent to which this type of space (i.e., Fig. 4) constitutes a satisfactory definition of the rollover index (or threshold) for a large variety of military vehicles as used by the U.S. Army.

7.0 RECOMMENDATIONS FOR FOLLOW-ON WORK

Possible further work is discussed below in two subsections, one in which improvements to the simulation techniques utilized in this study are proposed, the other in which the simulation of certain military vehicles of particular interest is considered.

7.1 Vehicle Simulations Employing HVOSM

The manner in which HVOSM has been used in this project has been found awkward in two respects. Firstly, relying on line printer output some time after a simulation run involves an inconvenient separation of question and answer, slowness in converging on solutions which require an iterative technique, and some difficulty in translating the numerical output into an accurate picture of the vehicle motions. Secondly, when the load to be carried by a vehicle is to be changed, the necessary long-hand data modifications are extensive. If significant further work with HVOSM were to be undertaken, we would propose that our graph plotting routines be employed to examine the results, in time history form, immediately after each run (on the screen of a Visual Display Unit), and that any results of lasting value be placed in the plotter job queue at that time. Also, if the effects of loading were to be studied extensively, we would propose automating the data modifications necessary to describe each new loading condition.

7.2 Simulation of Other Military Vehicles

Four specific cases, involving an M-151 towing a trailer, a three-axle medium truck, a small articulated vehicle, and a large articulated vehicle, should be studied. The existing HVOSM code is not suitable for simulating any of these vehicles or vehicle combinations. Developing HVOSM to make it suitable is not recommended since there are preferred alternatives. For example, the MVMA-HSRI simulation program is well suited to dealing with the last three vehicle types. On the other hand, no program is known to exist which will simulate the M-151 and trailer in rollover maneuvers. The best approach to simulating this last vehicle combination in limit maneuvering would appear to involve incorporating the independent rear suspension system as an option in the MVMA-HSRI program.

Data describing the trailer could be estimated relatively easily on the basis of dimensions and weights. Insofar as the military trucks are similar in design to non-military trucks, HSRI's experience in measuring the parameters of commercial vehicles and the existence of parameter data for many typical commercial vehicles should permit the estimation of parameters which would lead, through simulation, to a good understanding of their rollover behavior and problems, if any.

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